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HISTORY OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PRELIMINARY REPORT OF THE COMMITTEE ON SOCIETY HISTORY

CHAPTER VI

THE DEVELOPMENT OF THE LIBRARY

148 Among the several elements which largely contributed to the successful progress of the Society, the Library occupies a prominent position, and as it was largely due to the existence of the Library that plans for the acquisition of a permanent home for the association were effected, some account of its origin and development is appropriate in this place.

149 During the first two years of the existence of the Society no attempt was made to acquire books, and indeed by the close of the second year of its work it became a matter for serious consideration whether its work could be successfully carried on at all on the scale originally planned without some method of increasing its income. These early straits were successfully passed, however, and with the incorporation of the Society, and at the same meeting of the Council at which Professor F. R. Hutton was elected Secretary, the desirability of starting the acquisition of a permanent property in the form of a library was discussed, and the initial action to that end taken. To Mr. C. J. H. Woodbury is due the presentation before the Council of a plan of which the following abstract from the records of that body will give the substance.

150 At the meeting of the Council on February 15th, 1883, it was moved by Mr. Woodbury that the Secretary be instructed to

Under the direction of the Council, the Committee on Society History has arranged to present the results of its investigations to the members of the Society.

The Preliminary Report will appear in The Journal of the Society from month to month, and thus enable the matter to be open to comment during its completion. It is especially desired that any member who may be in the possession of facts or information bearing upon the various points as they are thus made public will communicate with the committee, in order that the final and completed report may have the advantage of the collaboration of the membership at large.

insert in the next communication to the members, and to the technical press, a request for circulars and price lists of manufacturing establishments and reports of engineering operations, with a view to making a catalogue of contemporaneous engineering work, to be properly filed and placed at the service of the members. In the price lists the ruling prices and discounts in January 1883 were requested to be affixed, such catalogues of machinery to be contributed as would show the growth and development of the industry to which they belonged. This motion, which was carried and put into effect, was the foundation of the present valuable library of The American Society of Mechanical Engineers, a library which is now housed, with the collections of the other founder Societies, the American Institute of Mining Engineers and the American Institute of Electrical Engineers, on the upper floors of the United Engineering Building, thus forming an integral portion of what is doubtless destined to become one of the great professional libraries of the world.

151 The response to the request contained in Mr. Woodbury's motion was prompt and liberal, many of the technical periodicals contributing complimentary copies of their publications, and some of them sending complete bound files of their back numbers. Manufacturers sent not only their trade catalogues but books to aid in founding the library. A standing committee on the library was appointed, and in the first announcement sent out by Secretary Hutton, dated March 1, 1883, the statement was made that the Secretary's office contained a growing collection of periodicals, transactions, and books accessible to members, and the hope expressed that in the near future the collection would receive such additions as would render it both interesting and valuable for reference.

152 At the annual meeting in New York in November 1884 the committee appointed to take steps for the definite organization of the library made an extended report, which will be found in full in the Transactions for that year. This report recommended the establishment of a permanent fund for library purposes and for the provision for its current expenses, that no demand upon the current funds of the Society need be made. Subscriptions for a permanent fund were solicited, and also contributions in the form of annual subscriptions of $2.00 or more, and an appeal was also made for contributions of books and papers relating to mechanical engineering. It is especially interesting to note the realization even at that early date that the library might become the incentive which should lead to the acquisition of a permanent home for the Society, and the following quotation

is given, as showing the beginning of an effort afterwards so abundantly realized:

"Accommodation for the Library to be provided in whatever rooms the Society may occupy. In this connection, however, your committee begs respectfully to call attention to the great desirability, for the advancement of the general interests of the Society, and especially for the adequate accommodation of the Library which it is hoped to create, of inaugurating early measures for the creation of a fund to provide a permanent building for the general uses of the Society."

153 Following upon this report of the Library Committee the Secretary issued a circular to the membership calling for subscriptions to the fund and to the annual contributing list. The result was that more than one hundred members responded, and the organization of the library was thus effected and continued, with modifications, until it was finally merged into that of the Society, as will be told hereafter. Reporting upon these facts at the Atlantic City Meeting, in May 1885, Mr. Henry R. Towne, chairman of the Library Committee, called attention to the desirability of providing accommodations for the Library, and mentioned the discussion of the construction of a union building for the several National engineering societies, showing the extent to which the idea had already taken root.

154 In the report of the Library Committee for 1885 was given for the first time a list of accessions to the library, and it is interesting to note that valuable books were contributed by members, while the exchange list included the principal technical papers then published in the United States and Great Britain, with some Continental accessions. These lists continued to be published in successive volumes of the Transactions, and showed a continual growth of interest in the development of the library, although the books were housed as yet in the limited quarters available in the Secretary's office where they were by no means convenient for general use.

155 Interest in the Library among the membership also began to show itself in the form of bequests and large contributions. Thus, in the report of the Library Committee for 1888 appears the bequest of the private library of Mr. Alfred B. Couch of Philadelphia, including a number of important and valuable books. In like manner, there was announced at the annual meeting in November 1889, an important gift of books from the library of the late Charles W. Copeland, formerly treasurer of the Society, the gift including many valuable books relating to the history and development of mechanical engineering. Important progress was also made in the completion of the files of the leading engineering journals, and the library began to assume real value as a reference collection apart from the important

part which it was soon to play as a financial asset in the development of the Society.

156 During the year 1890 there was brought up a plan for the acquisition of the house at No. 12 West Thirty-first Street, in New York, then occupied by the New York Academy of Medicine, the hall of which had already been rented for several meetings of the Society.

157 That the full advantages obtainable by corporations conducting public libraries might be realized, it was thought advisable to create a new corporation known as The Mechanical Engineers' Library Association, this to be a membership corporation of Fellows, consisting of the contributors to the library fund and governed by a Board of Trustees; both Fellows and Trustees being members of The American Society of Mechanical Engineers. This gave a special corporate body in which the title to the property might be vested to greater advantage than in the main Society and from which The American Society of Mechanical Engineers might rent the accommodations it required. Not the least of the advantages gained by the formation of this subsidiary organization was the exemption from taxation, belonging to public library corporations.

158 The books of the library continued to be the property of The American Society of Mechanical Engineers, being simply placed in the hands of the Library Association for the operative conduct of the library.

159 The development of the library fund during the years between its foundation in 1884 and the time of the purchase of the house in Thirty-first Street showed that it had attained the respectable sum of $2163.48, while the contributions to the current expenses of the library amounted to $1717.12.

160 The report of the Library Committee at the annual meeting of 1890 showed a marked increase in the number of periodicals on file while the transfer to the new quarters greatly increased the usefulness of the library by enabling it to be kept open in the evenings.

DISCUSSION

SURGE TANKS IN WATER POWER PLANTS

By Raymond D. Johnson, Published in June Proceedings

Mr. L. F. Harza I am interested in Mr. Johnson's paper, for the reason that I have spent considerable time during the past winter studying the same problem. I was, at that time, asked by Prof. Daniel W. Mead to make a mathematical study of the general problem of speed regulation of water wheels, to see if some much needed light could not be thrown upon this subject that would make the solution of speed regulation problems depend upon scientific analysis.

2 The studies were made for Professor Mead, and at his expense, for his forthcoming work, Water Power Engineering.

3 The problem of the standpipe arose in connection with the general problem of speed regulation, and the results accomplished were quite gratifying. So far as I knew at that time no other solution of the problem had ever been made. Apparatus has been built, and experiments are now being conducted by Mr. A. H. Ayers in the Hydraulic Laboratory of the University of Wisconsin to verify the formulae experimentally.

4 The portion of the paper relating to the differential regulator or the surge pipe and its use, is entirely new to me. Most of the equations, however, given by Mr. Johnson as applying to the simple standpipe, and some additional ones were derived by the writer by a different method, and two of his equations, [3] and [5], were obtained in much simpler form. The method used was as follows:

Let A = cross-sectional area of the penstock.
F = cross-sectional area of stand pipe.
q = water used by wheel.
v = penstock velocity.
y = instantaneous elevation of water in the standpipe above or below that in the forebay, or the "surge."

Y = maximum surge, neglecting friction and governor action.
T = time for return to normal level, neglecting friction and governor action.
T' = time for return to normal level, friction and governor action considered.
H = total head.
h = instantaneous head.
D_a = maximum upward surge when full load is rejected.
D_b = maximum downward surge below initial friction gradient.
D'_b = maximum downward surge below initial friction gradient with effect of friction omitted.
D_B = maximum downward surge below forebay level.

5 Now, it can readily be shown that

$$\frac{dv}{dt} = \frac{g}{L} \text{ (accelerating head)}$$

$$= \frac{g}{L} (y - \text{friction head in penstock})$$

Or

$$\frac{dv}{dt} = \frac{g}{L} (y - c\, v^2) \qquad [1]$$

It is evident that

$$\frac{dy}{dt} = \frac{A\, v - q}{F} \qquad [2]$$

6 If the governor keeps step with the change in head and maintains a constant power, then

$$q\, h = q_2 h_2$$
$$q\, (H - y) = A\, v_2\, (H - c\, v_2^2) \qquad [3]$$

7 Equation [2] now becomes:

$$\frac{dy}{dt} = \frac{A}{F} \left[v - \frac{v_2\, (H - c\, v_2^2)}{(H - y)} \right] \qquad [4]$$

8 Differential equations [1] and [4] form a complete expression of the existing relations, but are very difficult, if not impossible, to integrate mathematically. The integration has, however, been per-

formed for several problems by the successive application of the equations to small portions of the arc as described by Mr. Johnson, obtaining a damped harmonic similar to what he describes.

9 Attempts were also made by the writer to integrate equation [1] with the cv^2 term omitted, simultaneously with equation [4], thus to obtain the effect of governor action alone, and to integrate equation [1] together with [2], where q is constant and equal to Av_2, in order to find the curve resulting from friction alone. Both attempts have failed thus far.

10 If both friction and governor action are neglected, we have

$$\frac{dv}{dt} = \frac{g}{L} y \qquad [5]$$

$$\frac{dy}{dt} = \frac{A}{F} (v - v_2) \qquad [6]$$

11 The writer found these equations to resemble very closely the simple sine curve when plotted by the arithmetical method, and therefore assumed the equation of the general sine curve, differentiated it, and equated the derivatives to those given by [5] and [6] for certain determining points of the curve. By this method, expressions were readily derived for Y and T (one-half the wave cycle) and the following v–t and y–t equations obtained. Later on, the writer succeeded in integrating equations [5] and [6] mathematically, and obtained the same identical equations as before:

$$T = \pi \sqrt{\frac{F\,L}{A\,g}} \qquad [7]$$

$$Y = \pm \sqrt{\frac{A\,L}{F\,g}}\ (v_2 - v_1) \qquad [8]$$

$$y = Y \sin \frac{\pi}{T} t \qquad [9]$$

$$v = v_2 - (v_2 - v_1) \cos \frac{\pi}{T} t \qquad [10]$$

12 In order to obtain more valuable equations, which would include the effect of governor action and friction, the following method was used:

13 Let the time required to reach D_b and hence to reach approximately the value v_2, under exact conditions, be $\frac{T'}{2}$,

14 The time $\frac{T'}{2}$ will be slightly greater than $\frac{T}{2}$ when friction and governor action are involved, and the method of determining it will be given later (equation [22]).

15 It is evident that the number of foot pounds of energy which must be supplied by the standpipe in this time $\frac{T}{2}$ is equal to the energy required by the wheel plus that required to accelerate the water in the penstock plus that necessary to overcome the friction of the penstock minus that supplied through the penstock,

Or

$$E_s = E_w + E_a + E_f - E_p \qquad [11]$$

16 Now

$$E_s = w\, F\, D_b \left(H - c v_1^2 - \frac{D_b}{2} \right) \qquad [12]$$

where D_b is the maximum surge below the initial friction gradient for v_1, and is used in place of Y to distinguish it from the value obtained by the other formula.

17 Also

$$E_w = A\, v_2\, \frac{T'}{2}\, w\, (H - c v_2^2) \qquad [13]$$

18 And

$$E_a = \frac{w}{2g}\, A\, L\, (v_2^2 - v_1^2) \qquad [14]$$

19 To obtain E_f we have

$$d\, E_f = A\, v\, w \times c v^2\, dt \qquad [15]$$

where c is the friction coefficient and v is obtained from equation [10].

$$v = v_2 - (v_2 - v_1) \cos \frac{\pi}{T'}\, t$$

20 The integration of [15] between the limits $t = \frac{T'}{2}$ and o, gives

$$E_f = A\ w\ c \left[\frac{v_2^3\ T'}{2} - \frac{3\ T'}{\pi} v_2^2\ (v_2 - v_1) + \tfrac{3}{4}\ T'\ v_2\ (v_2 - v_1)^2 - \frac{T'}{6} (v_2 - v_1)^3 \right] \quad [16]$$

21 Also to find E_p we have

$$d\ E_p = H\ A\ w\ v\ dt,$$

where v is obtained from equation [10] as before. Integrating between the limits $\frac{T'}{2}$ and o, gives

$$E_p = H\ A\ w\ T' \left(\frac{v_2}{2} - \frac{v_2 - v_1}{\pi} \right) \quad [17]$$

22 Combining and simplifying:

$$D_b^2 - 2\ (H - cv_1^2)\ D_b = -\frac{2\ A}{F} \left\{ \frac{L}{2\ g} (v_2^2 - v_1^2) + C \left[-\frac{3\ T'}{\pi} v_2^2 (v_2 - v_1) + \tfrac{3}{4}\ T'\ v_2\ (v_2 - v_1)^2 - \frac{T'}{6} (v_2 - v_1)^3 \right] + \frac{H\ T'}{\pi} (v_2 - v_1) \right\} \quad [18]$$

$$D_s = D_b + cv_1^2 \quad [19]$$

23 The upward surge can be found by the same equation by a proper change of signs, but is unimportant since it is always less than the downward surge D_s for the same change of velocities.

24 If friction be omitted and T' be changed to T for reasons mentioned later, equation [18] reduces to

$$D'^2_b - 2\ H\ D'_b = -\frac{2\ A}{F} \left\{ \frac{L}{2\ g} (v_2^2 - v_1^2) + \frac{H\ T}{\pi} (v_2 - v_1) \right\} \quad [20]$$

25 To derive an equation for the maximum upward surge, D when full load is rejected, we may equate the original kinetic energy in the penstock to that expended in friction plus that used in raising

water in the standpipe. The energy lost in friction is found from equation [16] by putting $v_2 = o$
or

$$E_f = \frac{A\, w\, c\, T\, v_1^3}{6}$$

26 The other quantities are evident. This gives

$$\frac{W}{2g} \frac{A\, L}{} v_1^2 = \frac{A\, w\, c\, T\, v_1^3}{6} + \frac{w\, F\, D_a^2}{2}$$

Or

$$D_a^2 = \frac{A}{F} v_1^2 \left(\frac{L}{g} - \frac{c\, T\, v_1}{3} \right) \qquad [21]$$

27 Equations [18], [19], [20] and [21] are all theoretically exact, except for the assumption that the velocity-change takes place along a simple harmonic in time $\frac{T}{2}$. The true curve for a half cycle, as used, is scarcely distinguishable from a simple harmonic but its period or time for return of water in standpipe to normal level is greater than the value T, given by equation [7]. In two cases which the writer has solved by arithmetical integration and in the example given by Mr. Johnson of the conditions at the Feather River Plant, shown in Fig 2, the true value T' may be closely approximated by the following formula:

$$T' = \frac{D'_b}{Y} T \qquad [22]$$

where T is found from equation [7],

Y from equation [8], and
D'_b from equation [20].

28 The quantity T' is useful in itself as the true time for return to normal head, but its use in formula [18] for determining D_b is not advisable, as the writer has found by solving a number of problems that the value of D_b thus found, agrees almost exactly with the value of D'_b found from equation [20], in which equation the value of T from equation [7] is used. Equation [18] may thus be rejected entirely, and equation [19] becomes

$$D_B = c\, v_1^2 + D'_b \qquad [23]$$

29 For a basis of judging the accuracy of approximate formulas three problems have been selected, all of which have been solved by arithmetical integration, and whose correct solutions are therefore known. Problems 1 and 2 have been solved by the writer, and the solution by arithmetical integration of Problem 3 has been given by Mr. Johnson in Fig. 2.

	H	L	A	F	R	v_1	v_2	C
Problem 1.....	50	500	50.3	402.4	0.125	1.94	4.77	0.03
Problem 2.....	200	5 000	100	910	0.11	13	15	0.1
Problem 3	540	15 200	220	1964	0.112	9.8	11	0.1654

30 The degree of accuracy obtained in applying equation [22] is shown by the following table:

	T' (true value by arith. int.)	T' by equation [22]
Problem 1.....	40.6	40.4
Problem 2.....	153	156
Problem 3.....	240	234

31 The agreement of formulas ([18] and [19]) and ([20] and [23]) for D_B is shown below:

	D_B by equation [19]	D_B by equation [23]
Problem 1.....		4.66
Problem 2.....	28.2	28.85
Problem 3.....	26.7	26.0

32 This evidently indicates that equations [18] and [19] may be discarded.

33 The following comparison of the results of the writer's equation for D_B with Mr. Johnson's equation, and with the correct values, shows a remarkable accuracy of both methods.

	cv_1^2	By arith. integration	Johnson's equa. [7] $+ cv_1^2$	Writer's equation [23]
Problem 1.....	0.11	4.75	4.45	4.66
Problem 2.....	16.9	28.15	26.9	28.85
Problem 3.....	16.0	26.1	25.66	26.0

34 If full load ($v = 15$) be suddenly thrown off in problem 2, Johnson finds the maximum upward surge to be 43.5 ft. above pond level. The writer's equation [21] gives 48.8 ft., and the correct value obtained by arithmetical integration with one second intervals is 47.8 ft. This is the only problem which the writer has solved to check these formulas.

35 The above comparison shows that both methods for determining D_s give results very close to the truth, and much closer than the accuracy which can be obtained in estimating the maximum instantaneous load change which should be provided for. The danger of the piling up of waves also adds an additional factor which must be left to the judgment of the engineer.

36 In the opinion of the writer, the standpipe should not be built high enough to hold all the water when full, or nearly full, load is rejected by the wheels, but should preferably be built to overflow either at the top or through relief valves at the bottom set to waste when the water reaches a given height.

37 This overflow provision not only limits the upward surge but also thereby limits the maximum possible downward surge which can occur, and prevents the possibility of the piling up of waves above a certain value, which can be definitely figured, and depends upon the height of overflow above the forebay. The water which would waste at the rate intervals when full load is rejected would be of slight consequence, and the large additional height of standpipe, which would in many cases be necessary to conserve this water, would be uneconomical. For example, in problem 2 the standpipe would need to be built about 50 ft. above the forebay. The higher the standpipe is built the less often will it overflow, and economy of design would call for a proper balance between first cost and water economy. In addition to economy, the overflow serves to damp the surge and prevent the piling up of waves.

38 Mr. Johnson has based his original equations for the differential regulator upon some assumptions, the truth of which must be demonstrated by arithmetical integration or by experiment.

39 The writer has not yet had time to apply arithmetical integration to the differential feature, but as he claims to have done so many times himself, the form of pressure curve which he describes is doubtless correct. If the newly required velocity is to be obtained in approximately the same length of time as with a simple standpipe, it is evident that the average accelerating heads in both cases must be equal. Therefore, if the differential feature produces approxi-

mately a constant accelerating head, this head, and hence the surge, can be less than with the simple standpipe in about the ratio of the average ordinate of the sine wave to its maximum ordinate, or $\frac{2}{\pi}$ = 0.637. The differential feature doubtless changes somewhat the time required for this change of velocity, and hence also the relative amplitude of the surge waves.

40 Assuming that the surge is reduced by this differential feature, some obvious advantages result, as follows:

a When the unit has been working at part load, and full load is suddenly demanded, the reduction of the surge evidently helps to prevent the effective head from dropping to such a point that the power demanded cannot be delivered.

b The surge waves are made to die out more rapidly than with a simple standpipe, without overflow, which makes less likely the dangerous piling up of the waves from several successive gate movements.

41 To determine the general effect of the differential feature upon speed regulation, let us consider the duties of the governor. The speed of a unit will remain normal as long as the power output of the wheel equals the demand. This power output is represented by the quantity qh, where q is the actual water passing through the wheel, and h is the effective power head. The function of the governor is to maintain this power product at a value just equal to the demand by readjusting q for each change which may he made in the demand for power, or which may occur in the head, h.

42 Now, three influences oppose speed regulation:

a The governor cannot act until a change of speed has occurred, of a magnitude depending upon the sensitiveness of the governor. During this interval the wheels have been developing the original power corresponding to v_1, while the generator has been delivering the power corresponding to v_2. The result is the loss of, or absorption of, energy by the rotating parts, which results in a change of speed.

b After the governor has begun to move the gates, some time is required to complete the movement during which time the speed departs still further from normal.

c After the gates have reached their proper position, an appreciable time is required for the water in the penstock

to accelerate up to the newly required value. This time varies from zero, for a zero length of penstock and draft tube, to a period of many seconds for a long closed penstock.

43 The penstock should be measured, for comparing speed regulation conditions, from forebay or standpipe if one is used to tail water.

44 Now, if the turbines were to take their water directly from a simple standpipe, without appreciable length of penstock, then the first and second influences are the only ones opposing speed regulation. If we assume after the initial gate movement resulting from a load change that the speed returns to normal, then it will not again depart from normal except by an amount sufficient to actuate the governor. The slow wave-like variations which take place in h can be readily compensated for by the governor, without the sudden large gate movement which was at first here required to compensate for the load change. Poor speed regulation results from quick changes of effective head, rather than from large changes (within reasonable limits), and a small quick change in h may cause a larger speed variation than a much larger slow change.

45 In the case of the differential regulator the sudden increase of load is followed, as the gates move, by the sudden drop in effective head in the surge pipe, which in the standpipe, did not occur to an appreciable extent until the governor had made the gate adjustment for the load change, and even then took the form of a gradual instead of a sudden drop. Thus the surge pipe, in effect, adds to the load for which the governor must compensate. As the power of a wheel is proportional to the three halves power of the head, it follows that a sudden drop of 5 per cent in the effective head on a wheel would be equivalent in its demand upon the governor and therefore in its effect upon speed regulation to a sudden increase of load amounting to

$$\left(\frac{100}{95}\right)^{3/2} = 1.08$$

or 8 per cent. This must be added to the real load-increase to get the increase which is effective so far as speed regulation is concerned.

46 If there is a considerable length of penstock between the surge tank and the wheel, then the bad effect upon speed regulation is augmented. This is due to the fact that a smaller head than in a simple standpipe becomes available for accelerating the water in this portion of the penstock. A greater time is required to generate the

newly required penstock velocity under this reduced head, with a consequent deficiency of developed power and drop in speed of the unit.

47 The value which is placed upon speed regulation by different engineers and the attention which is given to this problem in the design of a plant differs widely. It is now, and must always remain, a matter of individual judgment as to how much in any particular instance can be spent to obtain close speed regulation. The effect of the surge pipe might not render the speed regulation unsatisfactory if the other factors controlling speed regulation were favorable.

48 The writer does not wish it understood that he is opposing entirely Mr. Johnson's differential scheme or trying to underestimate the value of the principle which he has discovered and developed. A knowledge of the exact effect of a resistance between penstock and standpipe is of much value, and much credit is due Mr. Johnson for his thorough analysis of the problem.

49 The writer believes that Mr. Johnson's criticism of Mr. H. E. Warren's able paper upon Speed Regulation of High Head Water Wheels is unwarranted. The governor controlled by-pass valve which Mr. Warren claims to be "the only complete remedy for the troubles in speed regulation caused by excessive inertia of a water column," is, in reality, the only scheme except the deflector nozzle, which can vary the power derived from a long penstock without change of effective head, and is therefore the most perfect scheme as regards speed regulation alone. Water economy, whether it results from the use of fly wheel, standpipe, or surge tank, brings with it speed regulation troubles, more or less severe, depending upon the natural conditions as well as upon the care and expense exercised in the design and construction of the plant.

PROF. I. P. CHURCH[1] The subject treated in this valuable paper is of such importance as to justify the devotion of a great deal of time

[1] The discussion here given is not the one read at the Detroit meeting. When the writer first received a copy of the paper, with the request for a discussion, only a few days remained before the date of the meeting; and being very much occupied, he informed the secretary that he would be unable to comply. A day later, noting what he thought to be a serious inadvertence on the part of the author of the paper, the writer forwarded a brief discussion which was read at the meeting. Subsequent correspondence with the author, however, showed that the language of the paper had been misinterpreted; and now, with an apology for previous hasty action, the present discussion is offered.

and attention on the part of hydraulic engineers and others. For his own part the present writer has been much interested in deriving numerical results in a few specific cases to ascertain the degree of approximation attained in the use of an important formula of the paper; viz: equation [7] of Par. 45. The laborious processes necessary to get anything approaching accurate results have taken so much of the writer's time, however, that he is obliged to limit this discussion to the consideration of the single relation referred to, and a presentation of the numerical results obtained from various data.

2 The fundamental relations between the quantities involved in the problem, as derived from the principles of mechanics, will first be presented. Fig. *A* shows, in a diagrammatic way, a vertical section of the long pipe, or main conduit, *E X*, and stand pipe (or "surge

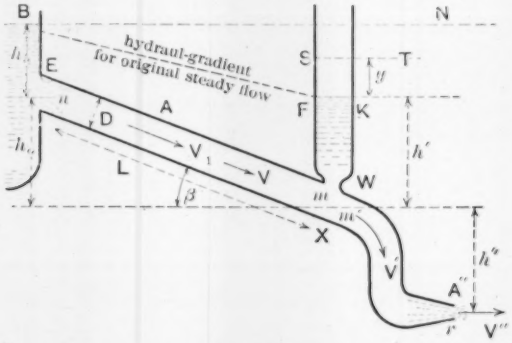


FIG. A SECTION OF CONDUIT AND STAND PIPE OF WATER POWER PLANT

tank") *F W*, etc., of a water power plant. As to notation, let *L* denote the length, *D* the diameter, and *A* the sectional area (i.e., $A = \pi D^2/4$), of the main conduit. The sectional area of the stand pipe is A/R, R being a ratio. For simplicity it will be supposed that the motor employed is an impulse wheel of the Pelton type, actuated by a "free jet" issuing from the nozzle at r; the sectional area of this jet being A'', and velocity V''. When normal steady flow is proceeding and the water in the stand pipe remains at rest, its level stands at *F K* and *B F* is the hydraulic gradient.

3 Let us now suppose that the "load" on the wheel is suddenly reduced below normal and is maintained constant at this lower value for an indefinite time; and that the governor and accessories are capable of maintaining a practically constant speed of wheel (equal to original) by varying the section of the jet, including a practically

instantaneous change at the very outset. An unsteady flow now ensues in the main conduit, with gradual reduction of velocity V of the water in it, from the original value V_1, as the water rises in the stand pipe. The velocity V' of the water at m', at the entrance of the short pipe $m'r$ (considered *to have the same sectional area at m'*, viz: A, *as the long conduit*) after an initial sudden diminution from its original value V_1 to a new value V'_0, will thereafter as the water rises in the stand pipe continue to diminish, but the diminution is gradual. Finally, when the rise y in the stand pipe has reached its maximum value y_m, V and V' will have diminished to a common value which may be called V_2. The analysis now to be presented will have to do *only with this first upward movement of the water in the stand pipe*, in the attempt to determine the value of y_m and (incidentally) V_2. The following six equations refer to any instant during this unsteady flow, the variables being V, V', V'', y, and the pressures p_n and p_m at points n and m. Fig. 4 shows the meaning of the symbols h_0, h', h'', h_n, and β. Time is denoted by t (an element of time by dt).

4 Although the flow is now unsteady, position n, just inside the entrance of the main conduit, is so close to the réservoir that no sensible error will arise in using Bernoulli's Theorem for steady flow between surface B and position n; whence

$$\frac{p_a}{\gamma} + h_n = \frac{p_n}{\gamma} + (1 + \zeta_E)\frac{V^2}{2g} \qquad [50]$$

where ζ_E is the "coefficient of resistance" for the entrance of conduit, p_a is atmospheric pressure, γ the weight of a cubic foot of water, and g the acceleration of gravity.

5 The inertia of the water in the long conduit EX is, of course, a very important element to bring into play. The net accelerating force acting on this cylinder of water, in the direction of its length, is $A\,p_n - A\,p_m + A\,l\,\gamma \sin\beta - \frac{f\pi D\,L\,\gamma\,V^2}{2g}$ where f is the "co-efficient of fluid friction." The mass accelerated is $AL\gamma \div g$ and the acceleration is $\frac{dV}{dt}$. Writing *accelerating force = mass × acceleration*, dividing through by $A\gamma$, and denoting $\frac{4fL}{2g\,D}$ by C_0, we find

$$h_0 + \frac{p_n}{\gamma} - \frac{p_m}{\gamma} = \frac{L}{g}\cdot\frac{dV}{dt} + C_0\,V^2 \qquad [51]$$

6 When the surface of the water in the stand pipe or surge tank is passing a position $S\ T$, y feet above the original, $F\ K$, the pressure at the base m must overcome the weight of the column of height $h'+y$, as also the atmospheric pressure p_a on the surface, and also accelerate ("overcome the inertia" of) the mass of that column with an acceleration $\frac{d^2y}{dt^2}$; and this leads to the relation

$$\frac{p_m}{\gamma} - \frac{p_a}{\gamma} = h' + y + \frac{h' + y}{g} \cdot \frac{d^2y}{dt^2} \qquad [52]$$

7 The pipe mr is supposed very short compared with the main conduit, and the flow from m' to r may hence be treated as a steady flow, justifying Bernoulli's Theorem; whence with ζ'' as the co-efficient of resistance for that pipe and nozzle, we have (neglecting V' as "velocity of approach")

$$\frac{p_m}{\gamma} - \frac{p_a}{\gamma} + h'' = (1 + \zeta'') \frac{V''^2}{2g} \qquad [53]$$

8 If Q cu. ft./sec. is the rate of flow through the nozzle at r, Q being also equal to $A''\ V'' = A V'$, we have, for the rising surge,

$$A\ V = Q + \frac{A}{R} \cdot \frac{dy}{dt} \qquad [54]$$

which may also be written

$$A\ V = A\ V' + \frac{A}{R} \cdot \frac{dy}{dt} \qquad [54a]$$

9 These are the fundamental relations needed for treating the problem, but it will be convenient to have also a relation holding good for the original steady flow, viz:

$$h_n + h_0 = h' + \left[\frac{1 + \zeta_E}{2g} + C'_0 \right] V_1^2 \qquad [55]$$

10 In the case of the long surge now under consideration the inertia of the water in the surge tank is of very slight consequence (as the writer has found by numerical trial) and hence in using equation [52] the term containing d^2y/dt^2 will be omitted.

11 To derive a relation between the three variables only y, V, and V', we combine equations [50] and [51] to get an expression for $\frac{p_m}{\gamma} - \frac{p_a}{\gamma}$

and then equate it to the right hand member of [52]; whence

$$h' + y = h_n + h_0 - \left[\frac{1 + \zeta_E}{2g} + C_0\right] V^2 - \frac{L}{g} \cdot \frac{dV}{dt} \qquad [56]$$

12 The coefficient (bracket) of V^2 is practically the quantity C used by Mr. Johnson, and called by him the "friction factor," and will now be written as such. Combining [56] and [55] gives

$$y - C\,(V_1^2 - V^2) = -\frac{L}{g} \cdot \frac{dV}{dt} \cdot \qquad [57]$$

But, from [54a], we have

$$dy = R\,(V - V')\,dt; \qquad [58]$$

and by multiplying [57] and [58] obtain

$$[y - C\,(V_1^2 - V^2)]\,dy = -\frac{RL}{g}\,(V - V')\,dV; \qquad [59]$$

which, in general, is a differential equation between the three variables y, V, and V'; (but V' is a function of y, due to the action of the governor, as will be seen).

13 During this first upward surge y varies from zero to the unknown y max. or y_m, and if we indicate the integration of the various terms in equation [59], just as it stands, denoting by V_2 the unknown common value of V and V' at the end of this surge, we obtain

$$\int_0^{y_m} y\,dy = -\frac{RL}{g}\left[\int_{V_1}^{V_2} V\,dV - \int_{V_1}^{V_2} V'\,dV\right] + C\left[V_1^2 \int_0^{y_m} dy - \int_0^{y_m} V^2\,dy\right] \qquad [60]$$

14 From the point of view of exact mathematics equation [60] cannot be completely integrated unless V' is given as a function of V, and V as a function of y. Both of these functions are unknown. But since V' is a known function of y through the nature of the regulating apparatus in this case of an impulse wheel, approximate methods are available, through equation [59], for finding y_m with sufficient accuracy for practical purposes.

15 Consider next the variation of V' as due to the action of the governor, in this specific case of an impulse wheel of the Pelton type. Let U = linear velocity of rim of the wheel (cup centers) at normal speed of wheel during the original steady flow, involving a definite value P (lb.) of the tangential "working force" acting on the cups. When the load on the wheel is diminished suddenly, the action of the governor will (instantaneously, say) so change the sectional area of the jet as to make the new value, P_0, of this working force smaller than P in the exact ratio of the reduction of load (otherwise the wheel would accelerate); and since the velocity V'' of the jet cannot increase until the surge begins to rise, this means that the rate of flow, Q, in the jet is suddenly made to take a value Q_0 cu. ft./sec. smaller than the original rate, and in proportion to the reduction of load.

16 From the mechanics of this impulse wheel, with e as its efficiency, we have just at the beginning of the surge, after the first quick action of the governor,

$$P_0, \text{ (lb.)}, = \frac{2e\, Q_0\, \gamma}{g}\ (V_0'' - U) \qquad [61]$$

where V''_0 = velocity of the jet at this initial instant.

17 At any later instant, the "load" (at its reduced value) being supposed to remain constant, and hence P_0 also, we have, similarly,

$$P_0 = \frac{2e\, Q\gamma}{g}\ (V'' - U) \qquad [62]$$

containing two variables, Q and V''.
Equating [61] and [62],

$$Q = \frac{(V_0'' - U)\ Q'_0}{V'' - U}\ ;$$

or, since $Q = AV'$, and $Q_0 = AV_0'$,

$$V' = \frac{(V_0'' - U)\ V_0'}{V'' - U}\ ; \qquad [63]$$

and this gives V' as function of V''.

18 Also, from [52] and [53] we have V'' as function of y, viz:

$$V'' = \frac{1}{\sqrt{1 + \zeta''}} \cdot \sqrt{2\, g\, [h' + h'' + y]} \qquad [64]$$

19 Equations [63] and [64] and the differential equation [59] afford the means of plotting curves, with fairly close approximation, showing

the variation of V' with y, of V with y, and of V' with V; and thus of determining finally the value of y_m for any given set of numerical data; and a check on the result consists in the application of equation [60] after the curves mentioned have been drawn. The determination of these curves involves lengthy arithmetical and graphic operations.

20 Mr. Johnson treats equation [60] by assuming V' to remain constant (sufficiently so), presumably at the value assumed by it at the beginning of the surge, thus making $V' = V_2$; and by integrating the last term in an approximate manner, which, perhaps, may be expressed as follows:—If the variable $C\ (V_1^2 - V^2)$ be denoted by Z, this last term may be written $\int_0^{y_m} Z dy$. When y is small Z is increasing slowly and dZ is less than dy; but later, as y nears its maximum, Z increases more rapidly than y, or $dZ > dy$, so that an approximate result may be reached by putting $dZ = dy$, as an average, throughout the whole summation; i. e.,

$$\int_0^{y_m} Z dy = \int_{Z_1}^{Z_2} Z\, dZ = \frac{Z^2}{2}\Bigg|_{Z_1}^{Z_2} =$$

$$\frac{C^2}{2}(V_1^2 - V^2)^2\Bigg|_{V_1}^{V_2} = \frac{C^2}{2}(V_1^2 - V_2^2)^2$$

And thus we are led to Mr. Johnson's equation [7], viz. (with present notation):

$$y_m^2 = \frac{RL}{g}(V_1 - V_2)^2 + C^2(V_1^2 - V_2^2)^2 \qquad [7]$$

in Par. 45 of his paper.

21 The writer will now give results that have been obtained by him in several numerical cases, beginning with those in which the design, while mechanically feasible, would be quite "un-practical" from an economical standpoint. It is thought, however, that the consideration of one or two cases of that character will lead to interesting and valuable mathematical indications. The units foot and second will be used.

22 Case 1. Let $L = 10\,000$ ft. and $D = 4$ ft., with $h' + h'' = 80$ ft.; and velocity V_1 of original steady flow in main conduit = 10 ft./sec. The diameter of the surge tank is taken as 12 ft.; hence $R = 1/9$.

Let the conduit be of riveted steel in which the loss of head, at 10 ft./sec., is as high as 11 ft. per 1000 ft. of length (authentic in one instance). With these data we find $\frac{RL}{g} = 34.50$ and $C = 1.11$. That is, in the original steady flow the velocity is $V_1 = 10$ ft./sec. in the main conduit, the total head is 191 ft. and the total loss of head is 111 ft. Let the velocity at m' change suddenly from 10 to 8 by the action of the governor and let it be supposed in this present case (1) to remain constant at that figure. This makes $V_1 = 10$, $V_2 = 8$; and $V' = V_2 = 8$, constant.

23 The curve for V as obtained by the writer from these data is shown in Fig. 1, and gives by its intersection with the horizontal through the value 8 of the vertical scale $y_m = 39.7$ ft. By measuring

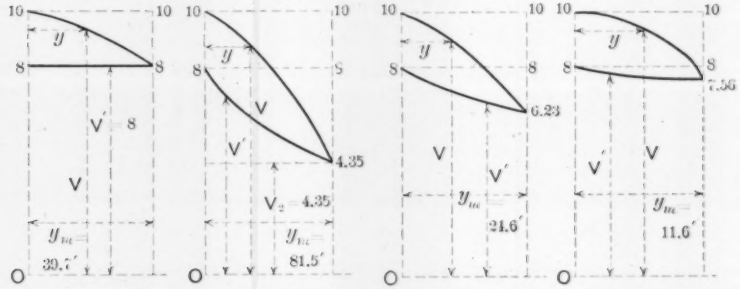


FIG. 1 FIG. 2 FIG. 3 FIG. 4

DIAGRAM ILLUSTRATING VALUES OF V

some ten values of V on this curve, properly distributed along the axis of abscissas, it was found that the mean V^2 was 83.56. In equation [60] we now put V' = constant = 8, $V_1 = 10$, and $\int_{V_1}^{V_2} V^2 dy = 83.56\, y_m$, etc.; that is,

$$\frac{y_m^2}{2} = -34.50\left[\frac{8^2 - 10^2}{2} - 8\,(8 - 10)\right] + 1.11\left[100 y_m - 83.56\, y_m\right];$$

or, $y^2_m - 36.50\, y_m = 138$; whence, solving the quadratic, $y_m = 39.95$ ft., a fairly close check.

24 For this Case 1, Mr. Johnson's formula [7] gives 41.6 ft., which

is seen to be only 4 per cent in excess of the other value derived by (fairly) rigorous methods.

25 Case 2. In this, and in the remaining cases, V' is to be variable, following the law of variation necessitated by the action of the governor in keeping the working force at rim of wheel constant at reduced load; as per equations [63] and [64].

26 Let the dimension values of L, D, and $h' + h''$, be as before, viz: 10 000 ft., 4 ft., and 80 ft., respectively; but for greater expedition in numerical work C is now taken a trifle smaller viz: = 1.00, and $\frac{RL}{g} = 30$ (which means a slightly larger tank section). The total head is now $80 + CV_1^2 = 180$ ft., and the loss of head = 100 ft. Take $V_1 = 10$ for original steady flow.

27 Now let the load on wheel be suddenly reduced by 20 per cent and hold the new value. The velocity V' will suddenly diminish from 10 to 8, the velocity of the jet being then, as during steady flow, of a value $V''_0 = 0.975 \sqrt{2g \times 80} = 70$ ft. / sec., and the (constant) velocity of the wheel cups one-half of this 70, or $U = 35$, which speed the governor will maintain. From equation [64] we have for any subsequent instant $V'' = 0.975 \sqrt{2g\,(80 + y)}$, while equation [63] gives

$$V' = \frac{(70 - 35) \times 8}{V'' - 35} = \frac{280}{V'' - 35}.$$

28 A curve being plotted for V' and y as coördinates, and also (the differential equation [59] being then brought into play) a curve for V and y, the intersection of these two curves (see Fig. 2) gives y max. as 81.50 ft.; and a value of 4.35 ft./sec. for the final, common, value of V and V' at the end of the surge. This 4.35 is the V_2 of equation [60].

29 From the curve for V and y, 63.40 is obtained as the mean of all the V^2's throughout the surge, with y as independent variable; while from the curve (now easily obtainable but not shown in Fig. 2) of V' and V the mean V' for V as independent variable is found to be 5.35. With these values placed in equation [60] and $V_1 = 10$, etc., there is obtained

$$\frac{y_m^2}{2} = 30\left[\frac{100 - 18.92}{2} + (5.35)\,(-5.65)\right] + 1.0\,[100\,y_m - 63.40\,y_m]$$

i. e., $y^2_m - 73.20\,y_m = 618.84$; and finally, $y_m = 80.85$ ft., as against 81.50 first obtained from the drawing.

30 If in applying formula [7] to this case we write $V_2 = 8$ the result is $y_m = 37.6$ ft. (54 per cent smaller than the 81.50); but if we write $V_2 = 4.35$ (although, as regards the use of [7], this value would be unknown in advance) $y_m = 86.7$ ft. is obtained, which is only 6 per cent in excess of the 81.50.

31 Case 3. We now take a somewhat more practical set of data, with $L = 5000$ ft., $D = 4$ ft., and a loss of head (for 48-in. riveted steel pipe) at the rate of 8 ft. per 1000 feet of length, with $V_1 = 10$ ft./sec., so that $C = 0.4$. Let the diameter of surge pipe be 16 ft. (nearly) leading to the value of 10 for $\frac{RL}{g}$. The whole head is 120 ft. up and the total loss of head is 40 ft. (which is still a large proportion, being one-third of total head); $h' + h'' = 80$ ft. and $V_1 = 10$.

32 As before, let the load suddenly diminish by 20 per cent, and remain constant at the new value so that V' suddenly changes from 10 to 8 ($= V'_0$). The two curves needed being drawn are found to intersect (see Fig. 3) at a point giving 24.60 ft. for y_m and 6.23 ft./sec. for the final V_2. The subsequent use of equation [60], with 74.52 as the mean V^2, and 6.81 as the mean V', gives a value of $y_m = 24.41$ ft., which is a close check on the above.

33 For the data of Case 3, if V_2 is considered as 8 in equation [7] the result of using [7] is $y_m = 15.7$ ft. (about 36 per cent too small); while if V_2 be taken as 6.23 (unknown beforehand, of course, in the use of equation [7]) a value of 27.2 ft. is found for y_m (about 11 per cent greater than the 24.6 above).

34 Case 4. In this case the data taken approximate to those of Mr. Johnson's numerical example in Par. 85. The total head is 225 ft. of which 25 ft. are lost in friction (here only one-ninth of total head), leaving $h' + h'' = 200$ ft. Length L = 5000 ft., $D = 6$ ft., that is, 72-inch riveted steel pipe, in which the loss of head is taken as 5 ft. per 1000 ft. of length when the velocity is 10. The diameter of surge tank is taken as 24 ft. (nearly) such that $\frac{RL}{g} = 10$; while with $V_1 = 10$ we have $CV^2_1 = 25$ ft.; i. e., $C = 0.25$. With a sudden 20 per cent reduction of load on the wheel, we have $V'_0 = 8$ ft./sec.; the load to remain constant at the reduced amount. The curves found for V' and V are shown in Fig. 4. The V curve is seen to be more "rounding" than in the other cases (and it should be said that in all these cases this curve must have a vertical tangent at the right-hand extremity, since at that point the value of $dV \div dy$ from equation [59] must be infinite, $V - V'$ being = zero). V_2, the final value com-

mon to V and V', is found to be 7.56; and $y_m = 11.55$ ft. With the value of 86.10 for the mean V^2 and 7.675 for the mean V', obtained from the proper curves, equation [60], after solution of the quadratic, gives $y_m = 11.6$ ft.

35 On substitution in equation [7] we find that if V_2 is considered to be 8, $y_m = 11.0$ ft., which is only 5 per cent less than 11.6; while if V_2 is taken to be 7.56 (the actual final value of both V and V_1) we obtain $y_m = 13.2$ ft. (some 14 per cent in excess of the 11.6).

36 By way of résumé of Cases 2, 3, and 4, it is seen that in using equation [7] if we regard the change of velocity occurring in the main conduit during this first upward surge to be in proportion to the reduction of load on the wheel (i. e., putting $V_2 = 8$ each time) the margin of error is small in 4, which is the most practical case; and very considerable in the other two, especially in Case 2, which is, however, a decidedly "un-practical" case. On the other hand, if for V_2 in [7] we substitute the actual final value of V in main conduit, only 6 per cent error is found in Case 2, 11 per cent in 3 (more practical), and 14 per cent in 4 (most practical). It need hardly be said, perhaps, that equation [7] was not intended to apply to such data (with relatively large friction-heads) as are found in Cases 1 and 2.

37 The writer has supposed a single Pelton impulse wheel to be in use in each of these numerical illustrations; not because that type of wheel might be considered advisable for heads of 80 to 200 ft., but because the data and treatment could thereby be made much more specific, as regards the action of the governor, than in the case of a turbine. Of course, with such a type of motor several wheels of the same radius and on the same shaft, or a few wheels with many nozzles, would be necessary to take care of the water in the foregoing instances. For simplicity the writer has considered only one wheel and one nozzle, but the result is the same.

38 Perhaps the foregoing numerical results may serve as rough guides as to what extreme change of velocity in the main conduit might advisably be assumed in using equation [7] in connection with surge tanks.

MORRIS KNOWLES Prior to the construction of the filtration plant, the water supply of Pittsburg was taken from the Allegheny River at Brilliant station, at the extreme northeast corner of the city. The water was taken through cribs about 150 ft. from the shore and pumped from open wells known as No. 1, 2, 3, 4, and 5, in the Brilliant Pumping Station. Wells No. 2, 3 and 4 were supplied through three

36–in. cast iron intake pipes and well No. 5 was supplied through a 48-in. cast iron pipe. These wells were all connected with one another and in this way the supply of well No. 1, which had no direct river connection, was taken from the other wells. The areas of the wells in square feet were as follows: No. 1, 153; No. 2, 325; No. 3, 184; No. 4, 326; No. 5, 154.

2 After the construction of the filtration plant water was taken from the filtered water reservoir on the other side of the river through two 72-in. mains, 1400 ft. long. Two methods were discussed for taking the water under the new conditions; one, to seal the wells and take water from open wells, the other, to use closed suction. The closed suction method was adopted as preferable to prevent contamination of the purified water. The 72-in. main delivered to an equalizing chamber, having an area of 136 sq. ft. located near the pumping station and from this a suction trunk led to the several wells where connections with the suction trunk were made to the pumping engines.

3 With the construction of the filtration plant, two new self-contained pumping engines of the vertical compound condensing, crank and flywheel, high duty type, of a capacity of 15 M. G. D. each, were installed in the pumping station. The pumping equipment, after this installation at Brilliant, is as follows:

1 10 M. G. D. pump, taking suction from wet well No. 1.
1 15 M. G. D. pump, taking suction from wet well No. 1.
2 15 M. G. D. pump, taking suction from wet well No. 2.
1 15 M. G. D. pump, taking suction from wet well No. 3.
1 12 M. G. D. pump, taking suction from wet well No. 3.
2 12 M. G. D. pump, taking suction from wet well No. 4.
1 10 M. G. D. pump, taking suction from wet well No. 5.
1 12 M. D. G. pump, taking suction from wet well No. 5.

These pumps are divided into two sets, one of which pumps to Highland Reservoir No. 1 with a hydraulic lift of 350 ft. including friction, the other to Highland Reservoir No. 2, which is about 100 ft. lower. The latter pumps are designed for the former lift and are so constructed as to work under the latter conditions without serious loss of economy.

4 While the pumps were taking their suction from the river, through the open wells, considerable surge was caused in the wells. This was particularly noticeable in wet well No. 4, from which two 12 M. G. D. pumps took their suctions. Normally the surge in this well measured 5 and 6 in., but upon synchronization of the two pumps, the surge increased to a maximum of 30 to 36 in. The surge also

tended to draw these pumps into step more often than the pumps drawing from the other wells. This condition, which existed when pumping from the open wells, showed the necessity of adopting some form of stand pipe or surge tank for use in pumping from the mains coming from the filtration reservoir.

5 The following table gives the data for the three sizes of pumps which were considered as affecting seriously the surge in a closed suction:

Description of pump	Angle of cranks	Strokes per min.		Length of stroke	Discharge gal. per stroke
		Max.	Normal		
10 M. G. D.	90 deg.	22.5	22	56 ft.	315
12 M. G. D.	90 deg.	20.5	20	62 ft.	417
15 M. G. D.	180 deg.	21.0	20	64 ft.	553

6 In calculating the size and location of stand pipes or surge tanks, curves representing the quantity of water pumped were plotted, using the time as an abscissa and the velocity of the plunger as ordinates; from these curves, the maximum requirement and duration of the same could easily be picked. By grouping these curves, the most satisfactory location of the stand pipe and the size necessary to supply the sudden demand of the pumps during maximum demand were arranged. Fig. 1 shows the arrangement of these stand pipes and the pumps taking suction from the various wells. Fig. 2 is a plan of the suction showing the connections and the location of the wells.

7 It was necessary in changing over from the river supply to that of filtered water, to connect one well at a time, not to interfere with the city supply. This arrangement gave opportunity to study the effects on the suction of the various pumps.

8 The first condition was with No. 1 well, connected up. The suction trunk was bulk-headed between wells 1 and 2. This arrangement connected one new 15 M. G. D. pump and one 10 M. G. D. When only the 15 M. G. D. pump was running, the surge in the tank was not noticeable. When the 10 M. G. D. was running alone, or with the 15 M. G. D., the surge amounted to 2 in. minimum, and a maximum of 4 in.

9 The second well was then connected, adding two 15 M. G. D. pumps but no extra area of stand pipe. This arrangement made no noticeable difference in the surge conditions, as we now had three

2-72 in. Pipe 1400 ft. long connecting to Reservoir. Capacity of Res. 50 M.G.
Gate Chamber
Equalizing Chamber Area = 126 sq. ft.
Gate Chamber
Elev. 58
96 in. Steel Main Laid in Concrete
Boiler Room Floor
Engine Room Floor
Stand Pipe 60 in. Open Top
36 in. Suctions to 15 M.G.D. Pumps
Suction reduces from 96 in. to 84 in.
12 in. Suction which supplies a 12 M.G.D. and a 12 M.G.D.
Suction reduces from 84 in. to 72 in.
72 in. Valve
36 in. Suctions supplying 2-12 M.G.D.
Stand Pipe 60 in. Open Top
36 in. Suction supplying a 12 M.G.D.
Stand Pipe 60 in. Open Top
30 in. Suction supplying 10 M.G.D.
Elev. 58
High Water Elev. 55
Elev. 50
Elev. 24.28
Elev. 2.00
40 ft.
30 in. Suction to 10 M.G.D. Pump
33 in. Suction to 15 M.G.D.
9 ft. 2 in.
12 ft. 0 in.
13 ft. 6 in.
12 ft. 0 in.
11 ft. 10 in.
11 ft. 10 in.
13 ft. 6 in.
12 ft. 0 in.
9 ft. 1 in.

FIG. 1 ELEVATION SHOWING ARRANGEMENTS OF STAND PIPES AND PUMP CONNECTIONS

30 in. Suction No. 1 Pump
36 in. Suction No. 4 Pump
Stand Pipe
33 in. Suctions.
No.3 and 5 Pump
72 in. Valve
12 in. Suction.
No.9 and 6 Pump
2 36 in. Suctions.
No.7 and 8 Pump
Stand Pipe
30 in. Suction No. 2 Pump
36 in. Suction No. 10 Pump
66 in. Main
6 ft. by 8 ft. Sluice Gate
272 in. Sluice Gates
272 in. Pipes from Filtered Water Reservoir
Equalizing Chamber

FIG. 2 PLAN SHOWING CONNECTIONS AND LOCATION OF WELLS

15 M. G. D. on the suction and the 10 M. G. D. pump (which was old and in bad condition at the water end) was not needed, and the result showed that practically no surge was noted in the tank.

10 The third condition was with the No. 3 well connected up. This added a new 15 M. G. D. and one 12 M. G. D., without any extra stand pipe area. With the four 15 M. G. D. pumps, the surge amounted to a minimum of one inch and a maximum of 3 in., depending on the number of pumps in step at one time. With the 12 M. G. D. pump by itself or in connection with any of the other group, the surge varied from 2 to 13 in. The hammer on the condensers was such, owing to this surge, that the tubing began to give way and had to be reinforced.

11 It was at this time that a paper on surge tanks contained in the June Proceedings of The American Society of Mechanical Engineers was published, and an endeavor was made to apply the formulae contained in this paper to the conditions experienced on this work.

12 The work done by Mr. Johnson in his article is in connection with the long sluggish waves caused by the sudden change of load, while the waves in this case are rapid and due to constantly changing conditions. In our first application of Mr. Johnson's formulae we assumed that the equalizing chamber would act as a surge tank and combined its area with that of the stand pipe, considering the same to be concentric at a point half way between the equalizing chamber and the end of the suction trunk. Under these conditions L would equal 1385 ft; $A = 28$; $R = 0.143$; V max. = 10 ft. 6 in. per second, and V average = 7 ft. 1 in. per sec. Assuming $C = 0.1$ and substituting in Mr. Johnson's formula, we find the maximum surge equal to 10 ft. 6 in. and the time equal to 33.4 seconds. As the maximum condition of draft due to the pumps on the suction trunk could last only 0.7 sec., the above value for surge is an impossibility, as the constantly varying draft would cause such interference in the surge-ways that the formula would not be applicable to these conditions.

13 In the second application, the equalizing chamber was considered to act as a reservoir with the one stand pipe as located 30 ft. from the equalizing chamber, and the length of the river crossing was not taken into consideration at all. The result of applying these conditions and dimensions to Mr. Johnson's formula was to secure a value with a maximum surge of 1.53 feet and a time of 1.12 seconds. Assuming an average period of draft to be 0.5 sec., we find that the wave has reached only part of its height or about 1.35 ft., when it is broken up by the reverse draft. Under the above conditions the

surge was measured in the stand pipe in wet well No. 1 and showed a maximum wave of $11\frac{1}{4}$ in. The time of oscillation in each of these tanks under the same condition was about 20 per min.

14 The No. 4 well was next connected with the suction, which added two more 12 M. G. D. pumps and one stand pipe. The surge was noted under various combinations of pumps running; with No. 3, 6, 7, 8 and 10, the surge was constant at 9 in. With No. 3, 5 and 6, 8 and 10 running, a maximum of 14 in., an average of 10 and a minimum of $5\frac{1}{2}$ in. were experienced; with No. 5, 6, 8, 9, 10 running at a maximum of 10 in., average 6 and mean of 2 in. were obtained. The observations were all taken in the stand pipe in No. 4 well every half hour for a period covering four days. Applying Mr. Johnson's formula to the latter of these three conditions and assuming the equalizing chamber as a reservoir and the area of the two stand pipes as combined and located 70 ft. from the equalizing chamber, the maximum surge would be 4.3 ft. and the time 2.15 sec. Assuming the period of maximum demand as 0.5 sec., the surge would be 1.03 in., and if we assume the period of maximum demand as 0.4 sec. the surge would equal 10 in., which agrees very closely with the conditions noted in the stand pipe.

15 This is as far as the work has progressed in the connecting up of the various suctions; work is now under way connecting No. 5 well, and another stand pipe will be located directly on the end of the suction trunk.

16 It was found upon adding the second stand pipe that the surge was practically all contained in this one stand pipe, as the water level in the No. 1 stand pipe was practically constant.

Mr. Chester W. Larner Mr. Johnson's paper is a very timely one and deserves the careful study of all engineers engaged in the development of water power.

2 The use of the standpipe for pressure regulations is, of course, by no means new, but its use for speed regulation is in a measure new, inasmuch as most standpipes have been installed with but little thought other than to provide against dangerous rise of pressure. Scientific pressure regulation is essential to good speed regulation but in general the regulation of pressure which has been thus far accomplished by the use of standpipes has not been productive of the best results possible in speed control.

3 This condition is attributable to neglect of the theory of standpipe action by those directly interested in their design. The problem

of proportioning a standpipe to the requirements of good speed regulation under a given set of operating conditions has received but scant attention. Now that the matter has had a public presentation it is to be hoped that it will receive the study which it deserves.

4 In my study of Mr. Johnson's paper I have resorted largely to arithmetical integration of the various pressure and velocity curves and the plotted results are presented herewith. They represent a large amount of work done in a very limited time, but I believe they are notwithstanding substantially correct.

5 By arithmetical integration I mean the arithmetical calculation of the successive values of the variables, at intervals of one second, through the cycle of change. In order that there may be no doubt or misconception regarding these results I shall describe the details of the work.

6 The curves presented are all for one problem and admit of direct comparison. The following values were used, utilizing the author's nomenclature except where otherwise specified:

$L = 5000$ ft. $A = 100$ sq. ft. $h = 200$ ft. $R = .227$. $C = 1/10$.

7 These values apply to the curves for the simple tank and also to the differential regulator, with the addition, in the latter case, of an internal surge pipe having an area of 29 sq. ft.

8 It should be noted that these values are the same as for the author's problem, Par. 85, etc., with the exception of R. His tank area is 410 sq. ft. (Par. 91) whereas mine is 439 sq. ft., making the area of the outer tank of differential regulator 410 sq. ft. I intended to make all values the same, but inadvertently fell into this error and overlooked it until I had gone too far to change. However, the difference is too small to affect the comparative results to any appreciable extent.

9 In applying the arithmetical process to the simple standpipe the results are unquestionably very accurate. The method is based on the assumption that at the beginning of each second there is an instantaneous change of level in the tank accompanied by an instantaneous change of velocity in the penstock. The variables are assumed constant during each second. In the case of an open tank they actually change very little during a single second and the process will produce very accurate results. The case of the differential regulator, however, is more complicated. There are more variables, all interdependent, and the changes are of much greater magnitude and of such a nature that they cannot be exactly computed, even for one second, by arithmetic.

10 In my calculations I have made the assumption that at the beginning of each second there is an instantaneous rise in the surge pipe (for rejected load) to the point where the head on the ports will produce a flow equal to the flow rejected (less that remaining in surge pipe), and that the inner and outer tanks then rise at the same rate during this second, thus maintaining a constant differential head.

11 Although I think this is the most accurate of several assumptions which can be made (and which I have tried) it is not exact. In order to determine its degree of accuracy I have developed the exact equation for these conditions, as follows:

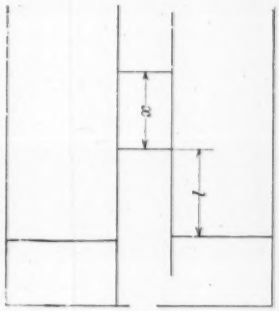


FIG. A

S = Surge pipe area.
l = difference of level between the two tanks at the beginning of any interval.
x = change of level in surge pipe during time t.
q = quantity of water entering or leaving standpipe during time t.
k = coefficient of discharge through ports.

12 Now equating the quantity entering, say, in an increment of time to that remaining in the surge pipe plus that flowing into the outer tank.

$$dt\, q = dt\, k\, a\, \sqrt{2g\,(x + l)} + d\,x\,S$$

$$dt = \frac{d\,x\,S}{q - k\,a\,\sqrt{2g} \times \sqrt{x + l}}$$

Let

$$k\,a\,\sqrt{2g} = b$$

Then

$$\int_0^t dt = S \int_0^x \frac{dx}{q - b\sqrt{x + l}}$$

Integrating and simplifying

$$\frac{tb}{2S} = \sqrt{l} - \sqrt{x + l} + \frac{q}{b} \log_e \left(\frac{q - b\sqrt{l}}{q - b\sqrt{x + l}} \right)$$

13 This equation holds for both demanded and rejected loads. It is assumed that q and l are constant during time t, and therefore the equation cannot be applied to any but short intervals during which these values are practically constant.

14 Unfortunately this equation is cumbersome and the value of x can be found by trial only. It is therefore not practicable to use it for determining x but it can be used to test values found arithmetically. For instance, I have taken the first second for curve Fig. 8. Here $q = 194$, $l = 0$, $x = 3.05$, $t = 1$. Substituting, the equation reduces to $4.95 = 4.98$, showing that the value of x found arithmetically was very nearly correct although a little large.

15 In my discussion I have treated four separate cases, Case 4 being the only one which represents practical operating conditions. The others are hypothetical and are of value only in the development and elucidation of the abstract principles involved. Their greatest value is in determining the separate effects of friction and governor action. Unless otherwise stated my reasoning applies to a demanded load.

16 Case 1 neglects the effects of both friction and governor action.

17 Case 2 neglects governor action but considers the effect of friction upon the velocity in the penstock. It neglects the effect of friction upon the quantity of water demanded by the wheel. This will be more fully explained later.

18 Case 3 neglects governor action but considers the *complete* effect of friction.

19 Case 4 considers both friction effect and governor action and is the case of an actual plant.

20 Now let us investigate in what specific way these four cases differ.

21 And here, since the author denominates his load changes in terms of ultimate velocity change, V_1 to V_2 or vice versa, let me

emphasize for fear of misunderstanding, that the wheel demand must be measured in *power*. It is $K q_2 h_2$ and is *constant* after the load change.

22 Now consider the case of a demanded load. After the load change the wave surges up and down in the standpipe and finally comes to rest at a level corresponding to the hydraulic gradient for the velocity existing in the penstock at that time. The wheel is getting the power which it demanded and conditions are quiescent. This ultimate or final velocity the author calls V_2. The head is the total head minus the friction head, or $h_2 = h - cV_2^2$. Also $q_2 = AV_2$.

23 As stated before, the demand of the wheel after the load change is a constant quantity; not a constant quantity of water but a constant quantity of power. The power *actually* supplied in Case 4 and the power *hypothetically* supplied in the other three cases is also *constant and equal to that demanded.*

24 The head after the surge has died out is less than it was before because the friction head is greater. It is $h - cV_2^2$ and before it was $h - cV_1^2$. It is quite evident then that under the higher head the wheels did not demand as much water as they do finally. The relation is

$$q_1 (h - cV_1^2) = q_2 (h - cV_2^2)$$

or

$$AV' (h - cV_1^2) = AV_2 (h - cV_2^2)$$

where q_1 is the quantity of water demanded at the beginning of the cycle, q_2 at the end, and V' is the velocity which will supply the demand q_1.

25 This relation is *entirely independent of governor action.* The governor takes cognizance only of drop of head in the standpipe while this relation exists *before* any drop has occurred and therefore before the governor has gone into action.

26 To make this equation general it is necessary to apply the aforementioned relation of equality between demanded and supplied power to all points of the curve. As will be shown later the author ignores this relation in the equations which he says include friction effect (corresponding to Case 2), and states, in correspondence with me, that I am in error in appying it to Case 3. His contention is that $q h$ cannot be constant without governor action and therefore that it is constant for Case 4 only.

27 I differ from him very decidedly. His statement is true only if we consider *actual* heads and in these three hypothetical cases we

cannot consider actual heads. We must consider the heads *assumed to exist under the hypothesis* and in that case $q_s h_s$ (supplied) is always constant and equal to $q_2 h_2$ (demanded). That relation must be true at all times because there is no other way to determine q_s except by its relation to h_s.

28 There are just two conditions which act to reduce the head and both appear in our hypotheses. It would stand at pond level in the tank if it were not pulled down by friction and by the surge. Since the surge has no effect upon the quantity of water demanded except through the governor, it follows that when we neglect governor action we do so by neglecting the surge.

29 In Case 1, by hypothesis, we neglect both surge and friction. Therefore by hypothesis the head is that which would exist if there were no friction or surge. That head is pond level or h.
Therefore from

$$q_s h_s = q_2 h_2$$

we have

$$q_s h = A V_2 h$$

and

$$q_s = A V_2$$

This is the quantity supplied to the wheel at any time. Since $q_s = AV'$, $V' = V_2$.

30 The author concurs in this and yet it depends on the relation which he says is not true.

31 In Case 3 by hypothesis we neglect the surge but consider friction. Therefore the head is that which would exist if there were no surge. That head is $h - c V^2$.
As before

$$q_s (h - cV^2) = A V_2 (h - c V_2^2)$$

$$q_s = A V_2 \left(\frac{h - c V_2^2}{h - c V^2} \right)$$

and

$$V' = V_2 \left(\frac{h - c V_2^2}{h - c V^2} \right)$$

The author, however, for this case makes $q_s = A V_2$ from which we would have

$$A V_2 (h - c V^2) = A V_2 (h - c V_2^2)$$

which is obviously false. He contradicts his hypothesis. It includes friction, and yet he takes pond level as his hypothetical head when the definite effect of friction upon the head is to introduce the gradient.

32 The author contends that in Case 3 the gradient does not exist but is obliterated in the surge. That is true. It does not *actually* exist but it *exists in the hypothesis.* For that matter neither does pond level, which he assumes for both of his cases, exist. It is obliterated by the surge just as much as the gradient is.

33 It is a fact that my Case 3 cannot be true without some movement of the wheel gates. They must open a trifle from V_1 to V_2. Exactly the same is true of his hypothesis however. If the wheels discharge AV_2 under $h - c\,V_2^2$ they cannot possibly discharge AV_2 under $h - c\,V_1^2$ unless the gate opening is changed. With constant gate opening q_s would vary as $\sqrt{h - cV^2}$. In either case it cannot logically be said that this condition involves governor effect upon the surge. That is the tendency which the governor has to exaggerate y, due to the surge, and we have neglected the surge.

34 Whatever view one may be inclined to take it is absolutely impossible to evade the point that whether governor action is considered or not the quantity of water demanded by the wheels during *the first instant* of the cycle multiplied by the head existing at that instant must equal $q_2\,h_2$. It makes it clearer to consider this first instant because it eliminates the drop in the standpipe, and with it any confusing thought of governor effect. The governor can have no effect until there has been a change of head.

35 I am devoting considerable space to this point because it is a very important one. It decides the question of whether an equation for practical use can neglect friction. Mr. Harza's equation [20] is such a case.

36 To recapitulate: for Case 1 the quantity drawn from the standpipe is at any time

$$Q_s = A\,(V_2 - V)$$

The accelerating head is $h_a = y$.

For Case 2 the quantity is the same as Case 1. The accelerating head is $h_a = c\,V_1^2 + y - c\,V^2$.

For Case 3:

$$Q_s = A\left\{V_2\frac{h - c\,V_2^2}{h - c\,V^2} - V\right\}$$

and

$$h_a = c\,V_1^2 + y - c\,V^2$$

For Case 4 we must consider the actual head and not the hypothetical head, and hence

$$Q_s = A \left\{ V_2 \frac{h - c V_2^2}{h - c V_1^2 - y} - V \right\}$$

and

$$h_a = c V_1^2 + y - c V^2$$

37 These expressions for quantity supplied and accelerating head define specifically the differences between these four cases. To integrate a curve arithmetically it is only necessary to figure the successive values of these variables one second at a time. The several figures show the pressure in the standpipe, the penstock velocity V and the "demanded velocity" V', all plotted to time in seconds.

38 An interesting relation between the pressure and velocity curves should here be noted. The quantity of water drawn from the standpipe in any increment of time is $\delta\, t\, A\ (V' - V)$. The total quantity in any time t is

$$\int_0^t A\ (V' - V)\, dt$$

In terms of y this is also $y \frac{A}{R}$. Hence

$$\int_0^t (V' - V)\, dt = \frac{y}{R}$$

But $\int_0^t (V' - V)\, dt$ is the shaded area between the two velocity curves. Therefore y is at all times proportional to this area.

39 In fact if t, $\frac{y}{R}$ and $V' - V$ were plotted to the same scale so that the product of one second into one foot velocity would be a unit of area then $\frac{y}{R}$ would be the integral curve of $V' - V$.

40 As it is I have measured all these areas with a planimeter and divided the area in square inches by the area of the rectangle representing one second time by one foot velocity and marked the result on each figure. It is, of course, in linear feet of flow, being the product of time into velocity. This area should be equal to $\frac{y}{.227}$ and it is in every case within a small percentage as any one can readily verify. These curves were largely figured on the slide rule which would account

for some error. This makes a very good graphical method for checking the accuracy of the calculations.

41 In the case of the differential regulator where tank level y' and riser level y are different the relation is

$$A \int_0^t (V' - V)\, dt = \frac{Sy + Ty'}{R}$$

where T is the area of the outer tank and S the area of the inner. When y and y' coincide it becomes the same as for the simple tank.

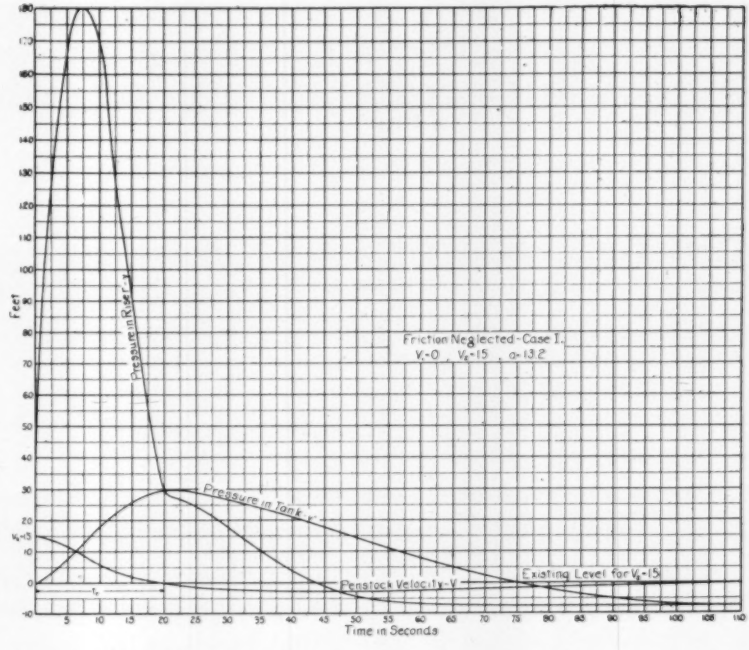

Feet
Pressure in Riser - y
Friction Neglected - Case I.
V₀=0 , V₁=.15 , a=1312
Pressure in Tank - y'
Penstock Velocity - V
Existing Level for V₁=.15
Time in Seconds

FIG. 1 CURVES FOR SHUTDOWN WITH DIFFERENTIAL REGULATOR—PORT AREA TOO SMALL FOR THIS CHANGE

42 Referring now to the figures, for a brief description, Fig. 1 shows the curves for a shut down with the differential regulator, *provided* the surge pipe is extended to prevent spilling. It does not represent an actual operating condition, because in practice the surge pipe would be cut off at less than half the height of the surge shown in this figure.

43 The curves were calculated principally to determine the characteristics in a case where the load change is much greater than that for which the ports are designed. They were designed, in this case, for a change from $V_1 = 13$ to $V_2 = 15$ (Par. 90).

44 The y curve rises rapidly and then falls rapidly until it crosses y^1, when the drop becomes more gradual. This peak is characteristic of all cases where the port area is too small for the load change.

45 Fig. 2 shows curves for a shut down with the port area increased to an adequate value. The value 36.8 was derived from the equation of Par. 90 (see also Par. 92). It seems about correct as y after reaching its maximum remains fairly constant. The first part of

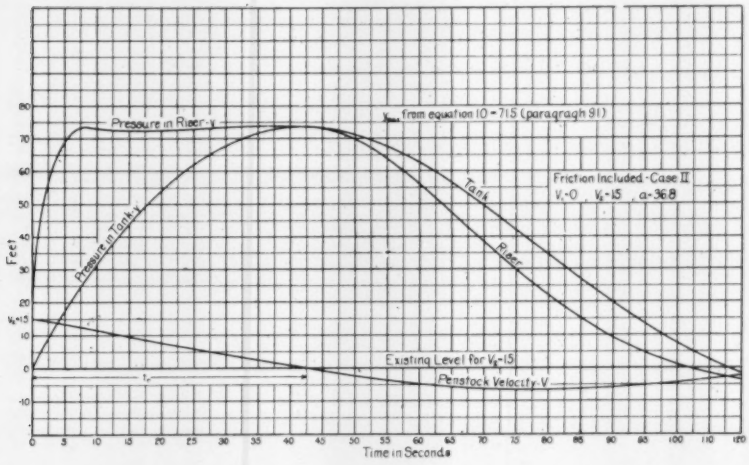


FIG. 2 CURVES FOR SHUTDOWN WITH DIFFERENTIAL REGULATOR—PORT AREA CORRECT FOR THIS CHANGE

this curve up to t_r, approximates the rectangular form which the author assumes in developing his equations for this type of tank.

46 This problem comes under Case 2 because the effect of friction acts upon V and not V', which is at all times 0. As has been stated the author's equation applies directly to this case and we should therefore find an agreement between the curve and the value of y_{max} which he deduces for these conditions. This is 71.5 ft. (Par 91) which practically checks the curve.

47 Fig. 3 is for a shut down with the port area about doubled. This was plotted to get the characteristics for the case of too large port area. I wished to see if the curve would not very closely approxi-

mate the curve of the simple tank. As can be readily seen it does closely approximate that shape of curve although the maximum surge is about 5 ft. less. The application of these data to operating conditions will be referred to later.

48 Fig. 4 was plotted for comparison with the author's problem in Par. 85. The maximum surge is 15.3 ft. whereas the author gives $y_{max} = 10$ ft. from equation [12]. That the discrepancy is due to the action of the governor can be seen by reference to Fig. 5 where I have plotted the same curve neglecting governor action. Here $y_{max} = 9.7$ ft. agrees pretty well with the author's value. The wide difference between these two curves should be particularly noticed

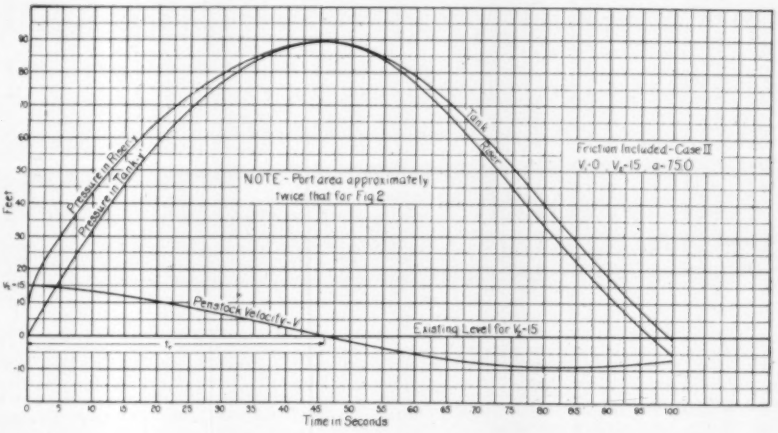


Fig. 3 Curves for Shutdown with Differential Regulator—Port Area too Large for this Change

in view of the author's contention that governor effect is not very important.

49 The shape of the y curve in Fig. 5 indicates that the port area is too large for this case. If it were smaller the curve would be more rectangular and y_{max} would be less, showing that the author's equation [12] gives values too large for Case 3. This is true. The equation can only legitimately be applied to Cases 1 and 2. If applied to Case 3 it gives values too large and if applied to Case 4 it gives values too small.

50 Fig. 6 and 7 show a corresponding comparison for the simple tank. Fig. 6 shows y_{max} for Case 4 to be 18.1 ft., whereas the value from equation [7] is 13.1 ft.

51 A comparison of Fig. 6 and 7 shows, as in the case of the differential scheme, that governor action increases y_{max} considerably. In this case it is from 11.6 ft. to 18.1 ft.

52 Fig. 7 also shows two dotted curves covering Cases 1 and 2 for the same conditions. Case 1 checks equation [2a] and Case 2 checks equation [7] as should be the case. These curves show that friction

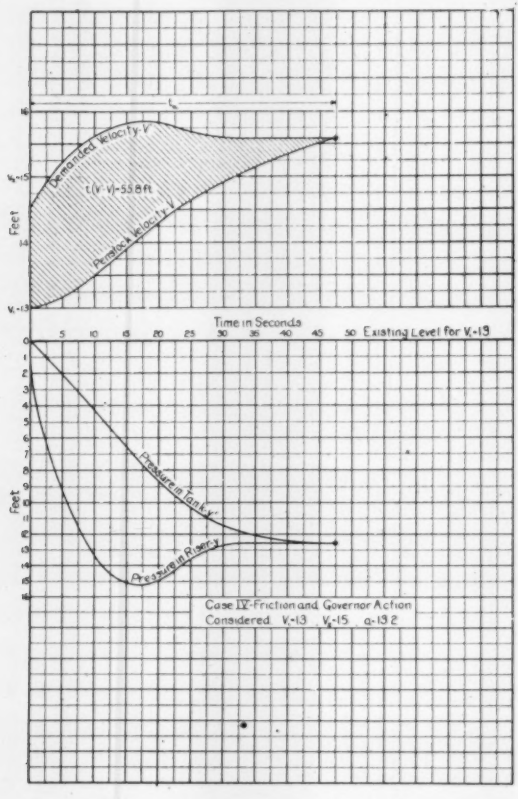


FIG. 4 CURVES FOR DIFFERENTIAL REGULATOR OPERATING UNDER CONDITIONS FOR WHICH IT WAS DESIGNED

has practically no effect upon y_{max}. In fact in this problem it reduces it a trifle; i.e., from 11.87 to 11.65.

53 I should mention here that this conclusion is entirely substantiated by the results of Mr. Harza's investigation. He has developed two equations for governor action, one of which [18] includes fric-

tion, whereas the other [20] does not, and he states in his discussion that "by solving a number of problems" he has found that the value of D_b (y_{max}) from equation [18] agrees almost exactly with the value found by equation [20] which indicates that the effect of friction is practically nothing. For this reason he rejects equation [18] in favor of [20].

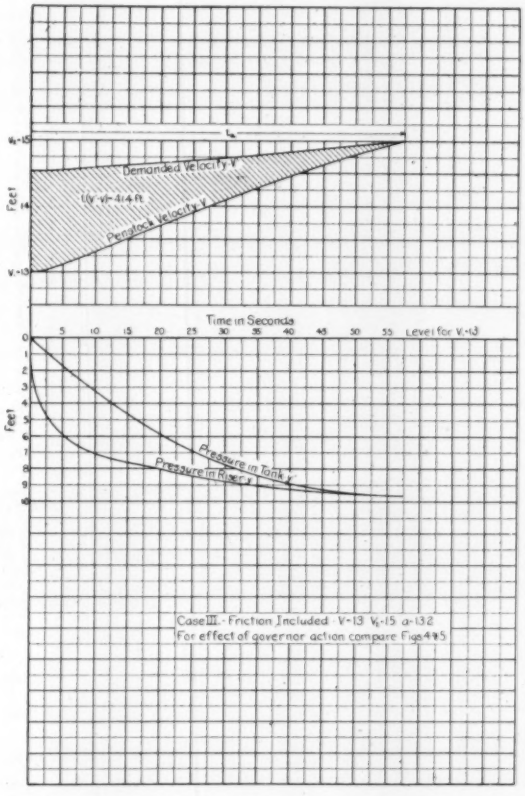

Demanded Velocity V
Penstock Velocity V
Feet
Time in Seconds
Level for V-13
Pressure in Tank y
Pressure in Riser y
Feet
Case III - Friction Included V-13 V1-15 a-132
For effect of governor action compare Figs 4 & 5

FIG. 5 CURVES FOR SAME CONDITIONS AS FIG. 4 WITH GOVERNOR ACTION NEGLECTED

54 Further than this, I have used his equation [20], which *neglects friction*, for my problem and get a value of 18 ft. for y_{max} and yet this value agrees with my curve Fig. 6 ($y_{max} = 18.1$) which *includes friction*. This certainly is further evidence that the consideration of friction has practically no effect on the surge. It does, however, increase t_a.

55 Fig. 8 and 9 show the effect of an increase of half a foot in the velocity change for both differential regulator and simple tank. These curves were plotted to ascertain the relative "elasticity" of the two types.

56 The effect upon the differential tank is much more marked than upon the simple tank. In the former case y_{max} goes from 15.3

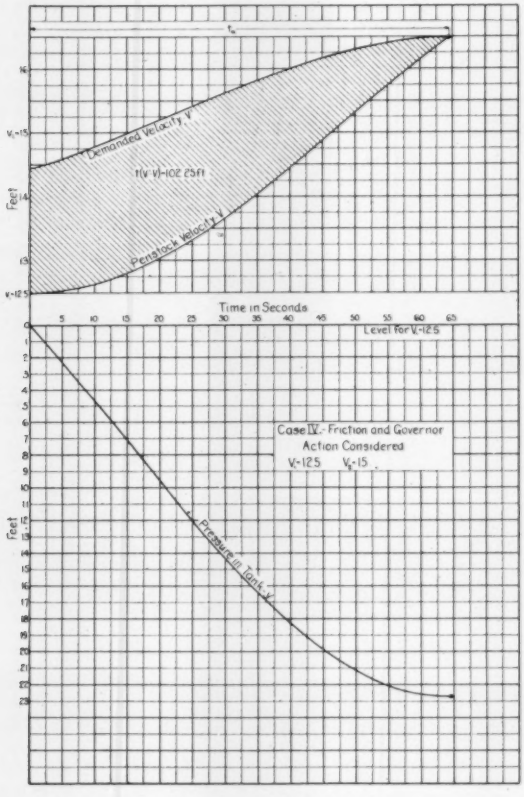


FIG. 6 CURVES FOR SIMPLE TANK OF SAME SIZE AS DIFFERENTIAL REGULATOR. SAME CONDITIONS AS IN FIG. 4

to 27 ft. owing to the change and in the latter case it goes from 18.1 to 22.7 ft. This shows that the differential type has considerably less "elasticity" and hence the importance of heeding the author's injunction to design for "the largest load change which can ever conceivably be suddenly demanded."

57 If the load change is increased a little more the differential tank will show a steadily increasing drop of such magnitude that the point is soon reached where the power demanded cannot be supplied and the limit of y_{max} will be the bottom of the surge pipe. Hence the absolute necessity in using this type of being able to predict the maximum load change.

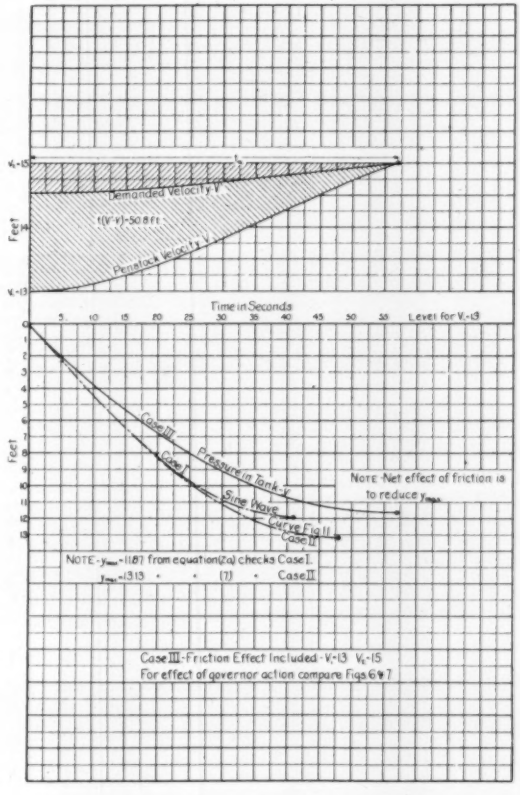


FIG. 7 CURVES FOR SAME CONDITIONS AS IN FIG. 6 WITH GOVERNOR ACTION INCLUDED

58 Fig. 10 and 12 show parallel curves for the two types under a reduced load change. This is a very important case because if we must design the differential regulator for the maximum load change it will be operating most of the time under conditions similar to those governing these curves.

59 The curves are for a velocity change equal to one-half the change for which the tank was designed. They show that under such conditions the y curves for both types are practically the same and that the differential tank loses its rectangular wave which is its mark of superiority. This was also shown by Fig. 3. This was a case of double port area which is equivalent to half load change.

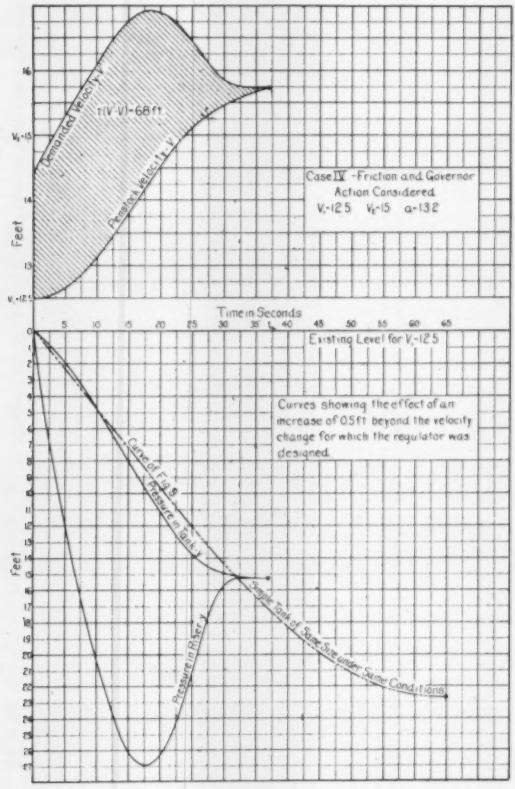

Demanded Velocity V
Present Velocity V
Case IV - Friction and Governor Action Considered
Feet
Time in Seconds
Existing Level for V₁-12.5
Curves showing the effect of an increase of 0.5 ft beyond the velocity change for which the regulator was designed.
Case of F.T.S.
Pressure in Tank y
Pressure in Riser y
Simple Tank at Same Size under Same Conditions

FIG. 8 CURVES FOR DIFFERENTIAL REGULATOR SHOWING EFFECT OF INCREASE IN LOAD CHANGE

60 Fig. 11 shows a complete set of curves for Case 2. These are for comparison with the author's friction equations.

61 Fig. 13 shows comparative curves illustrating the effect of varying load changes on both types.

62 The general conclusions derived from all the curves are:

a Friction has but little effect upon y_{max} in the case of the simple tank. An equation which includes governor action but neglects friction may therefore be safely used in practice. I have not investigated the case of the differential regulator to determine the effect of friction,

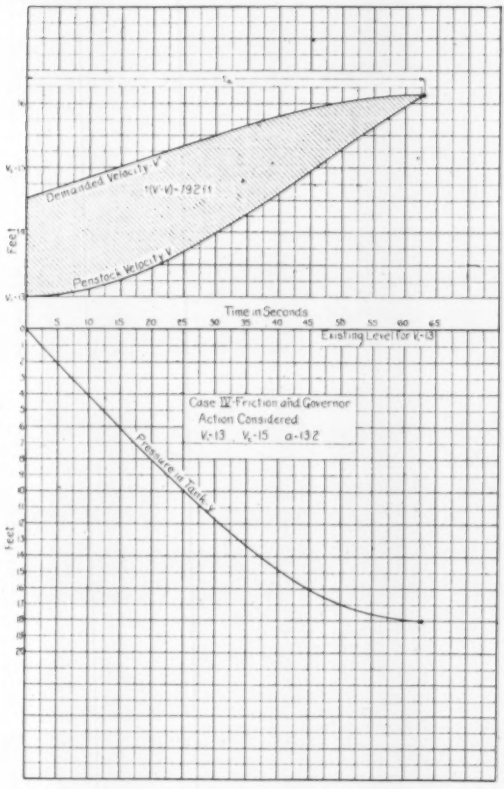


FIG. 9 CURVES FOR SIMPLE TANK UNDER SAME CONDITIONS AS IN FIG. 8

and am therefore not prepared to say whether it may be neglected or not.

b Governor action has a marked effect upon y_{max} in both types of tank.

c The differential tank has less "overload capacity" than the other. It will not operate satisfactorily under a load change much in excess of that for which it was designed.

d Under a change of about one-half that for which the differential tank was designed it has a pressure curve practically the same as a simple tank of the same size.

63 Referring now to the author's paper, let us first consider the equations for the simple tank neglecting friction and governor action.

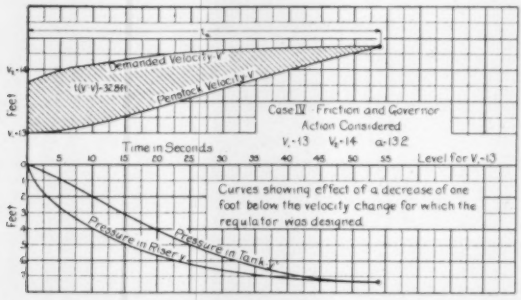

Demanded Velocity V
Penstock Velocity V
Case IV. Friction and Governor Action Considered
Time in Seconds
Level for V=1.3
Curves showing effect of a decrease of one foot below the velocity change for which the regulator was designed
Pressure in Riser y
Pressure in Tank y'
Feet

FIG. 10 CURVE FOR DIFFERENTIAL REGULATOR SHOWING EFFECT, OF A DECREASE IN LOAD CHANGE

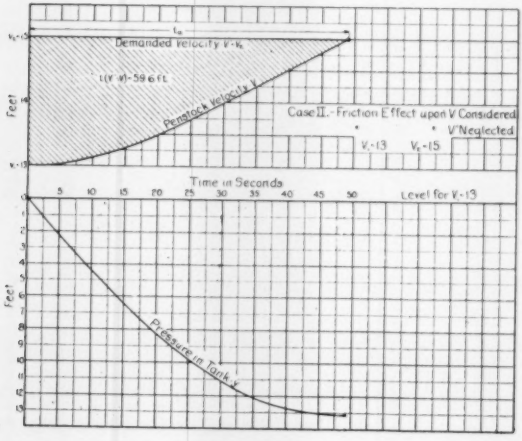


FIG. 11 CURVES FOR SIMPLE TANK

The fundamental differential equation of Par. 38 is the same as my equation for Case 1 (Par. 72). Anyone having any doubt as to its correctness can apply it to the sine wave shown in Fig. 7, which was integrated arithmetically, and verify the values derived therefrom.

64 The author introduces friction effect into this equation (Par. 43) by substituting $y - Z$ for the retarding head in the place of y.

In so doing he limits the application of this equation to Case 2 because he neglects the friction effect upon V'.

65 As stated previously, y is at all times proportional to the area between the curves V and V'. Hence the value of y is dependent upon both V' and V whereas this equation states that the value of y

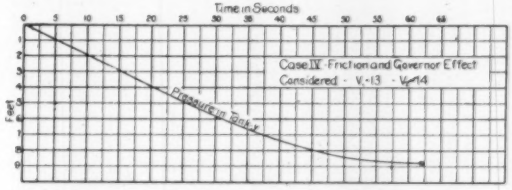


FIG. 12 CURVE FOR SIMPLE TANK UNDER SAME CONDITIONS AS IN FIG. 10

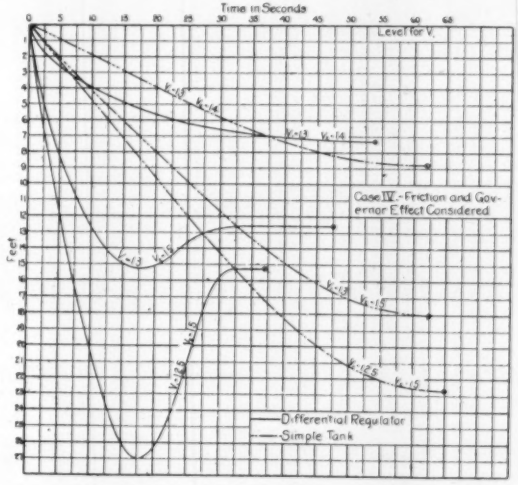


FIG. 13 COMPARATIVE CURVES FOR BOTH TYPES SHOWING EFFECTS OF VARIATION OF LOAD CHANGE

is dependent upon V alone, y and V being the only variables in the equation. It applies to Case 2 because there $V' = V_2$, a constant.

66 I am unable to perceive the logic of developing an equation for Case 2 because it is neither one thing nor the other. It neither neglects friction entirely nor considers it completely. I introduced it into this discussion merely for comparison with the author's equations.

67 That equation [7] is correct for Case 2 is shown by Fig. 2 and the dotted curve Fig. 7.

68 The true differential equation between y and V covering all cases is developed as follows:

$$\frac{dy}{dt} = R\,(V' - V)$$

also

$$\frac{dV}{dt} = \frac{g}{L} h_a$$

Combining,

$$\frac{dy}{dV} = \frac{RL}{g}\,(V' - V) \div h_a$$

To apply this to the several cases substitute for V' and h_a their respective values.

69 For Case 4 we would have

$$\frac{dy}{dV} = \frac{RL}{g}\left\{ V_2 \frac{h - cV_2^2}{h - cV_1^2 - y} - V \right\} \div (y - cV^2 + cV_1^2)$$

If this expression can be integrated it will give the true equation between y and V for actual operating conditions.

70 For Case 3 we would have

$$\frac{dy}{dV} = \frac{RL}{g}\left\{ V_2 \frac{h - cV_2^2}{h - cV^2} - V \right\} \div (y - cV^2 + cV_1^2)$$

71 For Case 2, putting $h_a = y - Z$, as the author does,

$$\frac{dy}{dV} = \frac{RL}{g}\,(V_2 - V) \div (y - Z)$$

or

$$dy\,(y - Z) = \frac{RL}{g}\,(V_2 - V)\,dV$$

which is the equivalent, for a demanded load, of the author's expression (Par. 43) for a rejected load.

72 For Case 1 we would have

$$\frac{dy}{dV} = \frac{RL}{g}\,(V_2 - V) \div y$$

and

$$\int_0^{y_{max}} y\, dy = \frac{RL}{g} \int_{V_1}^{V_2} (V_2 - V)\, dV$$

Integrating,

$$y_{max} = \sqrt{\frac{RL}{g}\,(V_2 - V_1)}$$

which is the author's equation [2a].

73 It is interesting to note, in view of the author's contention that Case 3 includes governor action, that the equation for Case 3 may be reduced to that for Case 1 by merely making $C = 0$; i.e., neglecting friction. It is also interesting to note the simplification of Case 4 if friction is neglected, particularly in view of the fact that friction appears to be negligible.

74 Putting $C = 0$ the equation becomes

$$\frac{dy}{dV} = \frac{RL}{g} \left\{ \frac{V_2 h}{h - y} - V \right\} \div y$$

Then

$$\int_0^{y_{max}} dy = \frac{RL}{g} \int_{V_1}^{\frac{V_2 h}{h - y_{max}}} \left\{ \frac{V_2 h - V h + V y}{h y - y^2} \right\} dV \qquad [1]$$

I have so far been unable to integrate this equation, but no doubt it can be done. The resulting expression would be a practical working equation for the simple standpipe from which the maximum surge for any tank could be determined directly from V_1 and V_2.

75 It should be here noted that the influence of governor action in any case will be great or small depending upon the extent to which y affects the expression

$$\frac{h - c\, V_2^2}{h - c\, V_1^2 - y}.$$

If h is large and y is small the effect of governor action may be inconsiderable.

76 The Feather River plant, for which curves are shown in the author's Fig. 1, is, I believe, such a case. Here $h = 540$ and $y = 10$. In this case y is so small relatively to h that its effect upon the value of the fraction is hardly appreciable. This would account for the fact that equation [7] practically checks the curve. At the same time it explains why equation [7] will not check any of my curves. In my problem $h = 200$ and y runs as high as 28 ft.

77 The error in the application of equation [7] to Case 3 is graphically represented in Fig. 7 by the area above the curve V' which is cross-sectioned in the opposite direction. Equation [7] makes y proportional to the area between V_2 and V whereas it should be proportional to the area between V' and V. This is only a rough illustration. It is not exact because the V curves in the two cases are not exactly the same. It serves to illustrate graphically, however, that the values from equation [7] are too great for Case 3. A comparison of Figs. 6 and 11 will show that they are too small for Case 4.

78 Equation [8a] for the differential regulator is the familiar $F = m\,a$, expressed in terms of t and V. It is likewise limited to Cases 1 and 2 in that it takes no account of V'.

79 It is further limited by the assumption that y is constant. In that case the pressure curve must have a rectangular form somewhat as shown in the author's Fig. 1 and in my Fig. 2. This limitation must be rigidly observed or results will go far astray. This can be easily demonstrated with any of the curves I show which are not of the rectangular form. As an extreme illustration take Fig. 1. The actual surge is 180 ft. whereas the surge from equation [10] is 71.5 ft.

80 The extent of the error in applying equation [12] to Case 4 is illustrated by Fig. 4. The curve shows $y_{max} = 15.3$ ft., whereas the equation gives 10 ft. The wave is not rectangular and hence some difference should be expected. Perhaps an increase of port area might improve it but the port area used was that determined by the author in Par. 90, for these same conditions, $V_1 = 13$, $V_2 = 15$.

81 This point brings up the limitation previously referred to. The equation cannot be true unless the curve is approximately rectangular and the curve cannot be approximately rectangular unless the port area is correct.

82 In Fig. 4 it looks as if the port area should have been larger. But it was proportioned according to the author's equation

$$a = \frac{A\,(V_2 - V_1)}{k\sqrt{2g\,y}}$$

which fact may indicate that this formula ought to be modified for governor action.

83 The formula is not theoretically true although I am not prepared to say definitely that it will not give close values in practice. It is not theoretically true because the quantity drawn through the ports immediately after the instantaneous drop is not $A\,(V_2 - V_1)$ when governor action and friction are considered.

84 The theoretical equation for all conditions of demanded load is

$$V_2 A (h - c V_2^2) = (V_1 A + k a \sqrt{2g y}) (h - c V_1^2 - y)$$

This equation is merely an expression of the condition that $q h$ demanded $= q h$ delivered. It will give the exact value of a necessary to start the pressure curve on its horizontal course, assuming an instantaneous drop y without acceleration of V_1. This is a consistent assumption, since it was used for the fundamental equation, and yet it would seem to make a a trifle large because in practice the drop is not instantaneous and there is some acceleration of V_1.

85 Applying this equation to the problem I find $a = 15.8$ instead of 13.2. This value might bring the curve of Fig. 4 down to a straight line but I have not had time to plot the curve to see.

86 In presenting my curves I have made comparisons between my results and the values obtained from the author's equations by inserting the same constants which I used for the curves. I did this in some cases to which the equations are not supposed to apply for the very purpose of emphasizing in a practical way the fact that they cannot be so applied. I also had the idea that, having used practically the same constants the author used in his problem in Par. 85, it would be possible to make comparisons.

87 I assumed that, having measured his load changes up to that point in terms of *ultimate* velocity change (Par. 35), he was doing the same in this case, i.e., that 13 ft. and 15 ft. velocity, which he gives as his limits, were V_1 and V_2 respectively. He states in his letters, however, that such was not his meaning; that 13 ft. is V_1, but that 15 ft. is the maximum *momentary* velocity during the period of acceleration and that, as per Par. 55, it is necessary to use this value in equations [7] and [12] in place of V_2 in order to approximate the surge with governor effect. This change of meaning of the symbols is somewhat confusing but if such is the case it is of course true that my curves do not check his problem. They do, however, check the use of the equations without modifications.

88 I notice that Mr. Harza in his discussion has taken these velocities for V_1 and V_2 and made comparisons between his equations and Mr. Johnson's on this basis exactly as I have done. I also notice that Mr. Johnson in Par. 91 uses these velocities as V_1 and V_2 himself to determine y_{max} for a shut down. This is a case *without* governor action and hence V_2 and not V_{max} is the value to use in the equation. But he puts 15 ft. in as V_2 which seems somewhat inconsistent. If V_{max} is 15 ft., then V_2 cannot be 15 ft. also.

89 I think that the natural conclusion of any one reading this paper with ordinary care would be that, given his constant S and V_1 and V_2, he could substitute them in the equations and get at least approximate results. I know I had that idea at first myself. It is true that the author states (Par. 55) that "it is best" to increase V_2 to "about"

$$\frac{h\,V_2}{h - y_1},$$

which is his value for V_{max}, but he does not say that this is *essential* in order to get an approximately correct result. In my problem it would seem that some such correction is absolutely essential although, as I shall show later, this is not the true value of V_{max} and if substituted for V_2 in equations [7] and [12] the values of y_{max} will be in error.

90 I wish here to state specifically that in the foregoing discussion I have not been attempting to show that the author's equations are of no value. I have simply tried to illustrate the discrepancies which exist *in this particular problem* between the true values and those derived from the author's equations *exactly as they stand.* I believe my comparisons are correct and will be of interest to readers who might be disposed to apply the equations, *just as they are,* to practical problems. At the same time I believe that if used by one familiar with their modification they can be made to yield practically correct values. But in this case equations [7] and [12] without modification show a considerable error.

91 Now before discussing the effect of substituting V_{max} for V_2 in equations [7] and [12] let us develop the true theoretical equation for the differential regulator including friction and governor effect and compare it with equation [12].

92 Following the method used for the simple tank:

$$\frac{dy'}{dV} = \frac{RL}{g}\,(V' - V) \div h_a$$

93 Under the hypothesis of an instantaneous drop in the riser and a constant value of y during the period of acceleration, V' becomes constant. We have

$$V' = \frac{V_2(h - c\,V_2^2)}{h - c\,V_1^2 - y}$$

and

$$h_a = y - c\,V^2 + c\,V_1^2$$

Substituting for h_a

$$\int_0^y dy' = \frac{R\,L}{g} \int_{V_1}^{V'} \left\{ \frac{V' - V}{y - c\,V^2 + c\,V_1^2} \right\} d\,V$$

Integrating and putting

$$b = \sqrt{\frac{y + c\,V_1^2}{c}}$$

we have

$$y = \frac{R\,L}{2\,c\,g} \left\{ \frac{V'}{b}\, log_e \frac{(b - V_1)\,(b + V')}{(b + V_1)\,(b - V')} - log_e \frac{y}{c\,b^2 - c\,V'^2} \right\} \qquad [2]$$

94 Given y, V_1 and V_2 this equation can be readily solved for R. It is theoretically exact under the hypothesis of an instantaneous drop and is therefore a practical working formula for an actual plant.

95 Combining the author's equations [11] and [12] and substituting V' for V_2 I find that the resulting expression is the same as my equation [2]. Since with the differential tank $V' = V_{max}$, it follows that by substituting V_{max} for V_2 in equation [12] the author will get true results *provided* he uses the *true* value of V_{max}. The value given in Par. 55 $\frac{V_2 h}{h - y}$ is not true when friction is considered. The true value is the same as that just given for V' in equation [2].

96 The effect of substituting V_{max} for V_2 can be demonstrated by reference to the figures.

97 For the differential regulator take Fig. 4. If the tank were so designed as to give the theoretical rectangular y curve then the curve V', starting from its initial value 14.5, would jump instantaneously to its maximum value $15.6 = V_{max}$ and then follow a horizontal course to the point of intersection with V.

98 The author's equation [12] is obtained by integrating the velocity change $V_2 - V$ between the limits V_2 and V_1. The area corresponding to this operation is that included between the curve V and the horizontal line $V_2 = 15$.

99 If V_{max} be substituted for V_2 the resulting equation will be the same as if $V_{max} - V$ were integrated between the limits V_{max} and V_1.

The corresponding area is that included between the curve V and the horizontal line $V_{max} = 15.6 = V'$. This is the true area and hence the substitution of V_{max} for V_2 in equation [12] will give true results.

100 For the case of the simple tank, refer to Fig. 9 in which the curves as drawn are theoretically as well as practically correct. In this case V' is variable and equals V_{max} only at the point where it crosses V.

101 The substitution of V_{max} for V_2 in equation [7] is equivalent to integrating $V_{max} - V$ between the limits V_{max} and V_1. This operation includes all of the area between the curve V and the horizontal line $V_{max} = 16.5$. The true area being that included between the curves V and V' it follows that the equation resulting from this substitution will contain an error represented by the area between the curve V' and the line V_{max}. Hence if the author makes this substitution, as he now claims to have done in Par. 85, and solves for his tank dimensions he will get too large a tank. The correct solution is the integration of $V' - V$ between V_{max} and V_1 (see my equation [1]).

102 Applying my equation [2] to the curve for the Feather River plant given by the author, where

$$L = 15\ 200 \text{ ft.} \qquad h = 540 \text{ ft.} \qquad V_1 = 9.8 \text{ ft.}$$
$$y = 7.5 \text{ ft.} \qquad c = 0.165 \qquad V_2 = 11 \text{ ft.}$$

we have

$$V' = \frac{11\ (540 - 20)}{540 - 15.8467 - 7.5} = 11.0713$$

$$b = 11.8952$$

$$cb^2 = 7.5 + 15.8467 = 23.3467$$

Substituting in [2]

$$7.5 = \frac{15\ 200\ R}{2 \times 0.165 \times 32.2} \left\{ \frac{11.0713}{11.8952} \log_e \frac{2.0952 \times 22.9665}{21.6952 \times 0.8239} - \log_e \frac{7.5}{3.1222} \right\} = 1430.45\ R \left\{ 0.921709 - 0.876367 \right\}$$

$$R = 0.115$$

103 The value of R used in plotting this curve by arithmetical integration is given by the author as 0.112. This is certainly as

satisfactory an agreement between theory and fact as could be expected.

104 Computing R for the same problem by the author's method we have (Par. 55):

$$V_{max} = \frac{h V_2}{h - y} = \frac{540 \times 11}{540 - 7.5} = 11.155$$

Also

$$Z = 11.895$$

Substituting in equation [11]

$$t_a = \frac{15\ 200}{64.4 \sqrt{0.165 \times 7.5 + (0.165 \times 9.8)^2}} log_e \frac{2.095 \times 23.050}{21.695 \times 0.740}$$

$$= \frac{15\ 200}{64.4 \times 1.9627} \times 1.10125 = 132.43$$

Substituting in equation [12]

$$7.5 = R \times 132.43 \times 11.155 - \frac{15\ 200\ R}{0.165 \times 32.2} log_e \sqrt{\frac{7.5}{7.5 - 0.165\ (11.155^2 - 9.8^2)}}$$

$$= 1477.26\ R - \frac{15\ 200 \times 0.489972}{0.165 \times 32.2} R = 75.49\ R$$

$$R = 0.099$$

This value of R is about 12 per cent low. The error is due to using the wrong value of V_{max} as previously explained.

105 To sum up in regard to the use of V_{max} for V_2, my conclusions are as follows:

a If the *true* value of V_{max} be used in the author's equations for the differential tank the resulting values will be correct.

b If the same modification is made in the case of the simple tank the results will be in error. The magnitude of the error will be in proportion to the variation between the initial value of V' and the maximum value.

106 If, as in the case of the Feather River plant, this variation is small then the error will be insignificant but not otherwise. For that problem $y_{max} = 10$ ft. in the simple tank and hence

$$V'_1 = \frac{V_2 (h - c V_2^2)}{h - c V_1^2} = 10.91$$

$$V'_{max} = \frac{V_2 (h - c V_2^2)}{h - c V_1^2 - y_{max}} = 11.13$$

The variation here is 0.22 ft. which is practically negligible.

107 As to the relative merits of the two types of tank from the standpoint of operation there is much to be said in favor of the differential scheme.

108 If used within its limits it will reduce the surge, as compared with a simple tank of the same size, and it has, at all times, more or less damping effect upon the pressure wave. This is an unquestionable advantage, especially in guarding against superimposed waves from rapidly succeeding changes of load.

109 It appears to have a disadvantage in the sudden drop of pressure at the beginning of the cycle but the examination of numerical examples shows that this drop is never really instantaneous and the disadvantage is only theoretical. The drop usually covers 10 sec. or more and any good governor can adjust the gates to take care of a large change of head in that time without the slightest difficulty.

110 If the port area could be made to vary with the load change it would be an ideal scheme. We could then have the rectangular wave for any change from zero to the maximum for which the regulator is designed.

111 Inasmuch as this is impossible it is well to understand that in actual operation the ideal condition will exist only for the change for which the regulator is designed. It has been shown that this type has very little overload capacity, for which reason we must design for the extreme load change. From which it follows that for ordinary changes, or normal conditions, we shall not have the ideal wave.

112 Suppose for example that the normal change is about one-half the maximum change. Then the pressure wave instead of being rectangular will be as shown in Fig. 10 which, as the comparison in Fig. 13 shows, is substantially the same as that for an open tank of the same size.

113 This, of course, is not at all to the disparagement of the differential tank. I merely desired to point out that we cannot expect to

realize the maximum benefit of this type for all variations of load change. Under the smaller changes it has practically no advantage but these changes are easy to govern and no advantage is needed. Under load changes which are hard to handle it has an advantage and ought to be the preferred type. It would be of maximum benefit in large plants where the sudden changes per unit are usually small. I agree with the author, however, that the greatest care is necessary in its design to insure satisfactory results.

CLUTCHES

By Henry Souther, Published in May Proceedings

Mr. Frank Mossberg Clutches for power presses must be so constructed as to disengage from the driving wheel and allow the driven shaft to stop in the same fixed position whenever disengagements are made. For quick running presses the clutch parts must be light enough to respond promptly when the trip lever is actuated, otherwise their inertia will so retard the action that the clutch will not properly engage with the driving wheel.

2 Perhaps the oldest clutch used for power presses is the simple jaw clutch illustrated in Fig. 1, in which the construction is so fully shown that no explanation is necessary. This clutch, while it may have proved useful for comparatively slow power presses and for light work, has been largely superseded by more improved types.

3 The principal objection to this clutch is its heavy parts, which make it slow to respond, and sometimes when the speed is up to 100 revolutions, difficulty is found in making the clutch jaws enter the driving recesses in the wheel.

4 Fig. 2 shows a form of clutch used largely by several Connecticut press manufacturers. This consists of a sliding key *B* fitted to move freely in a slot or pocket in the crank shaft, and controlled by a wedge shaped cam *A* connected to a treadle or hand lever. To lock the clutch to the driving wheel the cam is released and the spring forces the key into engagement.

5 Simplicity and low manufacturing cost are points in favor of this clutch, but it requires in practice considerable repairs.

6 Fig. 3 illustrates a form of clutch used extensively by the Stiles & Parker Press Company, and probably originated by our Mr. Stiles. It was developed some thirty years ago and yet is still used on a large number of presses where it appears to work satisfactorily.

7 In this clutch the locking pins *A* are located in the driving wheel, and held in position by the trigger block *B*. To operate the clutch the cam segment *C* is slightly depressed. When held in this position the trigger block located in the driving wheel will come in

contact with the cam lever, releasing the clutch pin, which in turn drops into recess *D* in the crank shaft. The cam lever is now returned to its normal position and when the shaft revolves the clutch pin *A*

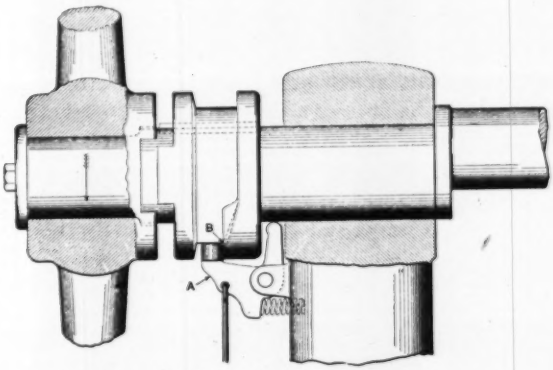


FIG. 1 PLAIN JAW CLUTCH

will be drawn out of the recess of the crank shaft by this same cam, thus stopping the driving shaft in its original position.

98 Fig. 4 shows a clutch used by one of the large press manufacturers, which commends itself for its simplicity. Owing to the

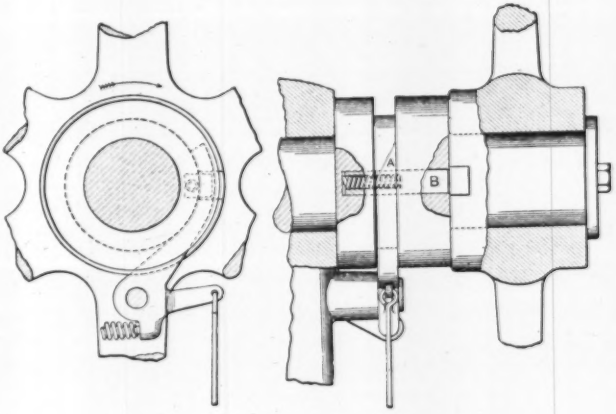


FIG. 2 SLIDING KEY CLUTCH

peculiar construction, however, whereby a considerable part of the bearing in the hub of the driving wheel is cut away, the bearing is liable to wear more rapidly. Its operation is as follows:

9 The lever *B* is swung out of contact with clutch arm *C*, allowing clutch arm and key to be acted upon by spiral spring shown dotted. The continual revolution of the driving wheel will promptly bring one of the recesses *E* in front of the half round locking key, allowing this key to make one-fourth turn in its sockets and thus lock the driving wheel to the shaft.

10 When the shaft reaches the original point of starting, the clutch lever *B* being in its normal position will obstruct the clutch lever *C* and cause the clutch key to return to its original position, allowing the driving wheel to pass over it freely.

11 Fig. 5 shows a form of clutch used on presses made by the Stiles & Fladd Press Company. The clutch pin *B* in this construction is placed in a pocket in the driving shaft in which it moves radially.

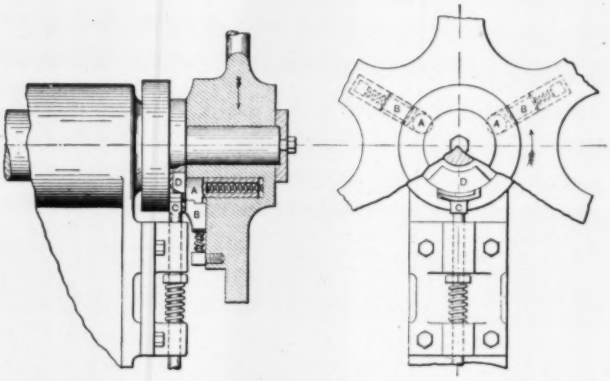


FIG. 3 STILES CLUTCH

A spring forces this pin outwards to engage with a lug in the driving wheel when the release lever is actuated. This actuating mechanism will also cause the clutch pin to recede into the pocket in which it is held by the releasing cam out of contact with the driving wheel. This clutch, having two or more contact points in the driving wheels, is a very satisfactory working mechanism and has proven very durable.

12 Fig. 6 shows a form of clutch used by the Ferracute Machine Co., and I think designed by Oberlin Smith. This clutch is used to a large extent on punching presses. It is rather complicated, especially in the form applied to larger presses. A desirable feature is that it is usually made with several contact points in the driving wheel

which are of tempered tool steel. These contact points are so constructed that when the clutch is engaged the wheel is locked in both directions. The illustration shows the working of this clutch.

13 Fig. 7 shows a form of clutch designed by the author and intended especially for large presses. It has two large engaging pins of tempered tool steel, which act together. These are located radially opposite each other and far enough from the center for good leverage. This clutch can be made to connect at each half revolution of the driving wheel. It is one of the strongest made and has proved serviceable in practice, requiring very little repairs.

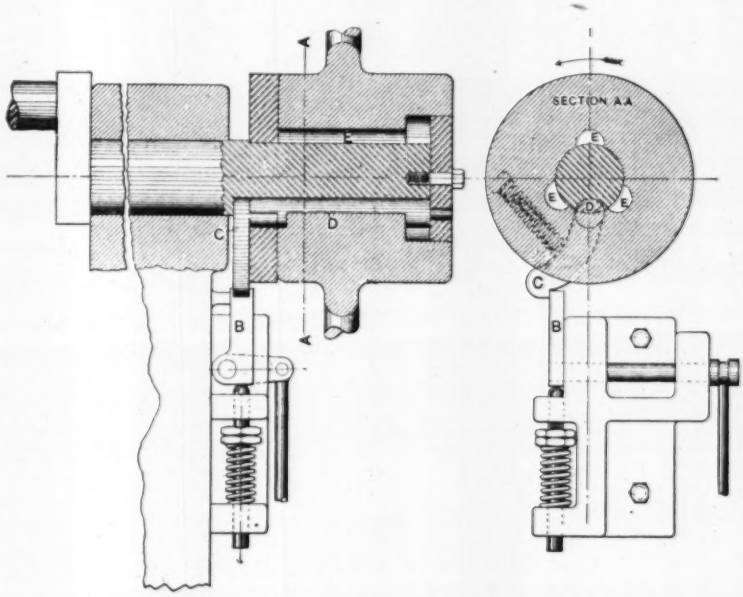


Fig. 4 Bliss Clutch

14 The clutch pins *G* connected with sliding collar *C* are mounted to move freely, and revolving with the driving wheel at all times. To operate the clutch, the locking pin *E* is pulled out off the cam *D*, allowing the collar *C* and pins *G* to move into engagement with the clutch lugs *H*, which lock the driving wheel and the shaft together. When the shaft returns to its original position the clutch pins are withdrawn by the action of the cam *D*.

15 In the foregoing I have covered briefly the construction of power press clutches in general use. There are many modifications

of these, but all of a class which makes a locking contact with the driving wheel and shaft at a fixed point or abutment in the wheel.

16 A total departure from all those previously described is a clutch invented by James A. Horton, of Boston, some years ago. Fig. 8 shows this clutch, which consists mainly of a hardened steel cam *A* keyed to the crank shaft; a clutch ring *C* mounted to turn slightly on the shaft; a series of rollers *B* held loosely in slots in said ring, and a spring *G* acting on the clutch ring and causing the same to turn, carrying with it the rollers *B* towards the high point of the clutch cam *A*.

17 The balance wheel is recessed to receive this clutch mechanism, said recess being lined with a hardened tool steel ring *D*. The diame-

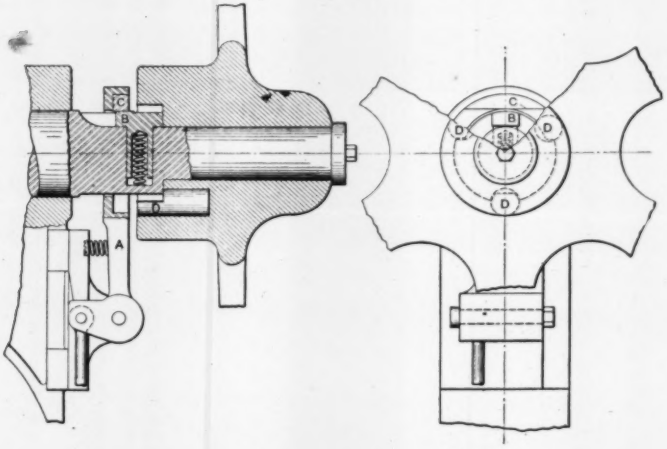


FIG. 5 STILES AND FLADD CLUTCH

ter of this recess is such that when the rollers *B* reach a point about half way between the lowest and highest point of the cam they come in contact with the clutch ring *D*, and act as wedges to lock the clutch. Release of the clutch is made when the lug *F* fastened to the clutch ring strikes against the stop lever *E*. This will throw the clutch rollers out of engagement and allow the wheel to pass freely.

18 This clutch I will term as a positive friction clutch: positive because it grips the load instantaneously when the clutch is actuated and carries it without any slip; friction because the contact between the driving wheel and clutch rollers is simply a smooth concentric surface.

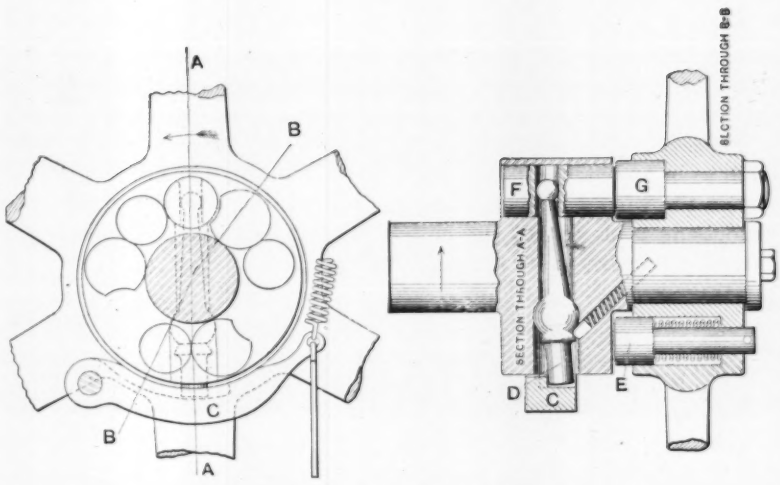


FIG. 6 FERRACUTE CLUTCH

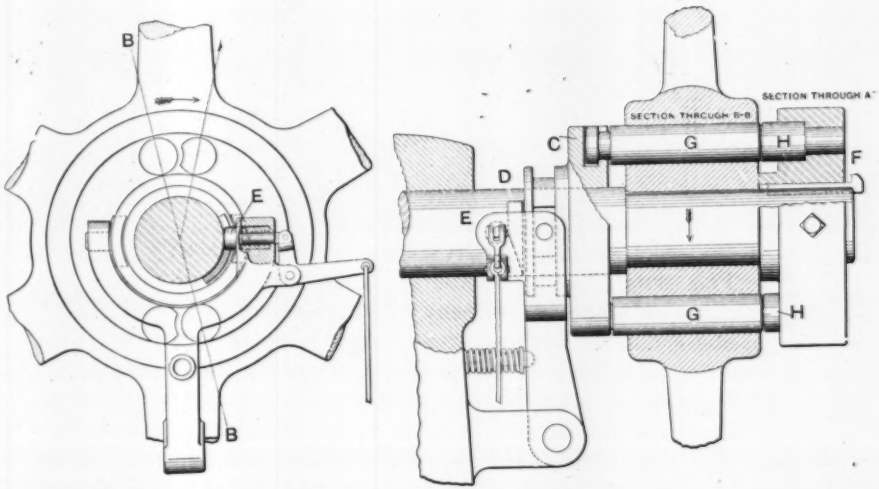


FIG. 7 MOSSBERG CLUTCH

As will be seen, in this clutch there are a number of locking points which act together and consequently the load is distributed. The clutch is instantaneous in its action and can readily be disengaged at any predetermined position on the shaft.

19 A desirable feature peculiar to this clutch is that it can be released with ease under full load. In other words, with this clutch it is possible to cause the slide of the press to descend on the work such as in embossing. Release the clutch when the embossing dies act on the work and then when the desired time has elapsed for the embossing tools to act the clutch is again engaged and press slide caused to return to its normal or up position.

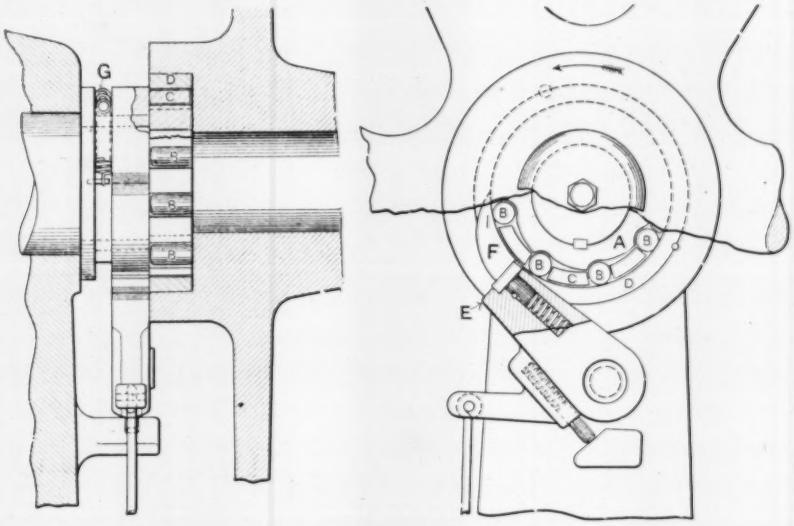


FIG. 8 HORTON CLUTCH

20 This clutch has been extensively used for power presses by the Standard Machinery Company of Providence, R. I., and is one of the most durable forms yet produced for this purpose. It is suitable for the lightest as well as the heaviest press made and works well for speeds from the slowest to 500 r.p.m. Expense of manufacture has probably alone hindered its more general adoption.

21 The instantaneous action of this clutch when the trip lever is actuated enables the operator to run a press fitted with it faster and keep more perfect time than with any other, as is shown by the following explanation of the working of the ordinary clutch previously described.

22 In the ordinary clutch the operator presses the treadle which actuates the fixed locking or catching key. The balance wheel is revolving around the shaft and may be just past the locking point, requiring almost an entire revolution to return to the fixed point in the wheel, while at another time the fixed point in the wheel is in such a position as to engage the locking point instantly; thus we have a stroke of the press taking practically two revolutions of the wheel in one case and only one in the next, this variation making it impossible to perform operations in unison of time.

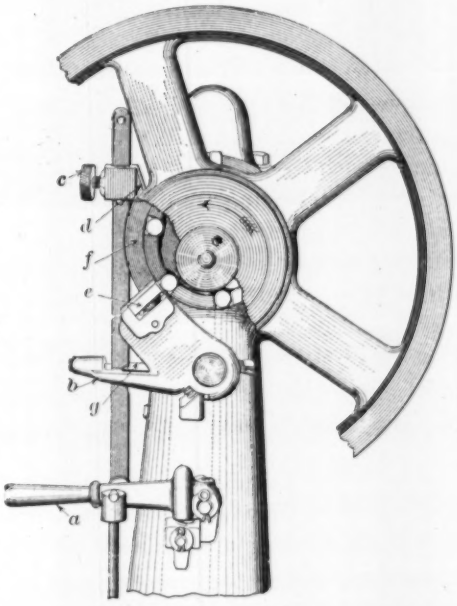


FIG. 9 HORTON CLUTCH APPLIED TO POWER PRESS

23 By reason of its instantaneous action, and the possibility of release at any point even under load, the Horton clutch is adapted for other purposes than power presses, as for automatic drop hammers, and is the only clutch that has come under the writer's observation that could possibly be used for the requirements of this machine.

24 The clutch is located in the hub of the driving gear and is used to engage the crank shaft with this gear. To operate the drop the hand lever is tripped, causing the hammer to drop and perform the work required, and immediately upon the rebound of the hammer

the clutch picks up the load without any loss of time and carries the hammer to the top position, when the clutch is disengaged and left ready for the next operation. These clutches are made in various sizes to transmit from ½ h.p. up to 1000 h.p.

25 Fig. 9 shows the Horton clutch applied to a punching press. It also shows the tripping levers and automatic safety device which guard against the press making more than one stroke at a time, excepting by tripping the starting lever for each stroke. To make the press run continuously the clamp *D* is raised to the top of the vertical rod shown and fastened with a thumb screw *C*. The operation and function of the mechanism is so plain that detailed explanation is unnecessary.

26 The safety device described is only one of many for use in connection with a press and a press clutch and by no means confined to the Horton clutch. The high speed at which small hand operated presses are run makes such a device very desirable for the protection of the operator's hands and fingers. Without it a press will frequently make a second stroke unexpectedly, when the operator is putting in or removing from the dies the piece which has been operated upon, and this is the time when accidents usually happen. Small presses run at a high speed, and fed by hand, especially, should be provided with clutches that will not start the press until the operating lever is tripped.

STRAIN ON PRESS CLUTCHES

27 Press clutches must endure exceptional strain and abuse. Often a press with its heavy fly wheel will be brought up standing by the operator's placing the work in the die in such a way that the press cannot make the full stroke. With a fly wheel weighing seven or eight hundred pounds and 30 to 40 in. in diameter the strain on the clutch is enormous; but a clutch that will not stand this abuse occasionally without breaking is not considered desirable or practicable. Several of the clutches described are capable of meeting these conditions.

MR. CHARLES D. RICE Fig. 1 shows a friction clutch of a general plan that I have come to regard as the most reliable of all with which I have had to do; the frictional part consisting of a proper quality of hard wood set on end in receiving pockets, the exposed ends being forced against a smooth metallic surface of conical form.

2 Another refinement which I have not seen before in practice is that the ends of the actuating fingers are supplied with rolls, afford-

ing great releasing force with slight effort by the lessening of friction at their contact with the wedge cone. The fingers also possess counterbalancing features that they may not hold the clutch from disengagement in running at very high speeds, when applied in reverse form to what is common in automobile practice.

3 The adjusting ring or nut which affords a seat for the wedge fingers to heel against also affords a universal adjustment for the whole, rather than the unbalanced adjustment common to some other forms of clutches wherein there are three or more members to adjust; this ring nut being split at one point and clamped at its thread by means of a screw (not shown).

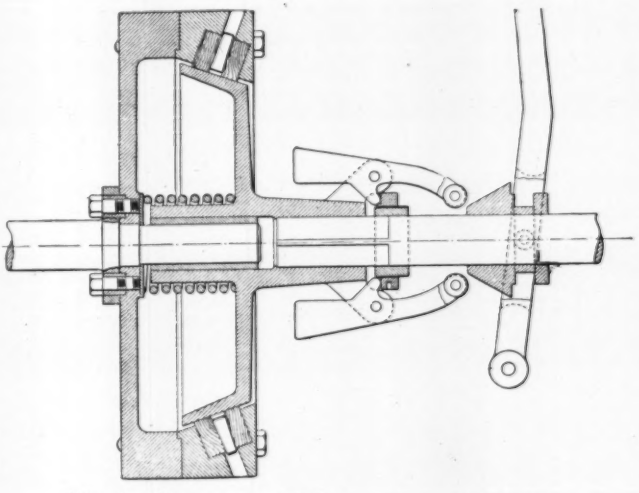


FIG. 1 CLUTCH WITH WOODEN FRICTION BLOCKS

4 The wooden plugs or settings are from well-seasoned hard wood, preferably maple, boiled in oil; they are wedged in their respective positions by means of a round tapered pin which can be easily removed. Practice has proven that such a clutch will drive well when the frictional surfaces are liberally exposed to oil, enabling the clutch to start a load effectively and without undue violence while gripping.

5 Some years ago I had occasion to apply a friction clutch to a large press. The diameter of this clutch at the gripping point was 6 ft. The wood settings were substantially of the character described, and the duty of the mechanism was sufficient to require an eight-ton fly-wheel running at 100 r.p.m. The clutch was actuated several

hundred times each day and furthermore must make an effective engagement to meet with the full resistance of the work within a rotating movement of 60 deg. The gripping surfaces were constantly exposed to a drainage of oil leading from the fly-wheel hub. It was hydraulically actuated, and operated very satisfactorily for a term of years without need for repair.

MR. F. C. BILLINGS The author of the paper holds that the cone clutch, *properly mounted*, is, all things considered, the best form of clutch to use for automobile purposes.

2 We have built several automobiles at The Billings and Spencer Company, and have come to realize this very point. Absolute flexi-

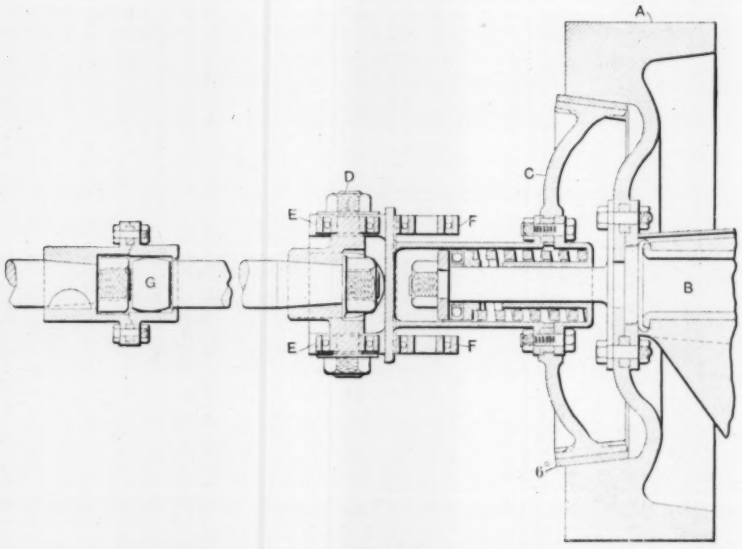


FIG. 1 CONE CLUTCH WITH FLEXIBLE MOUNTING

bility back of the cone is essential to its good behavior, and we went to extremes in our last car to obtain this flexibility. The above drawing will make clear how this was worked out.

3 Referring to drawing: *A* indicates the cast iron fly wheel, mounted upon crank shaft *B; C* the driven member of the cone clutch. The cone is leather faced with a 6 deg. angle, which is a smaller angle than any mentioned in the paper; but it is perfectly successful and does not wedge. At *D* is a universal joint of the usual cross type, but in place of bearing of hardened spindles or the like, four Hess-Bright

ball bearings are supplied as indicated at *EE*. These bearings naturally give perfect flexibility and freedom from binding in any direction.

4 We have overcome the tendency of the square telescope shaft to bind and stick at a critical moment by inserting two of the ball bearings at *FF*. These carry the torque with perfect freedom and give telescopic slip.

5 It is our belief that one universal joint is not sufficient, and we have therefore supplied at *G* a squared socket enclosing a nut with spherical faces as shown in the drawing. This permits full freedom of action for the universal joint ahead of it.

6 We realize that we adopted an expensive construction, but believe the expense to be worth while and that freedom from clutch trouble and absence of wear will compensate for it. A perfect acting clutch furthermore saves wear and tear on tires and the driving mechanism.

MR. FREDERICK A. WALDRON The writer was identified with the development of the multiple disk clutch, examples of which are presented by Mr. Souther, when first used on cranes in 1887–1890, and a summary of results may be of general interest.

2 A multiple disk clutch is a good clutch to hold but a poor clutch to release, for the following reasons:

a There is not enough lever movement to produce the clearance between the disks necessary to prevent dragging.

b As the disks are free on the shaft they are spaced more or less irregularly, causing rubbing of their surfaces. The clearance is greatest between the surfaces nearest the sliding part.

c Imperfect workmanship and heavy lubricants are a source of drag troubles, especially if the driven portion has a small static resistance when not in duty.

3 When the driving surface is moving continuously and the driven only part of the time, and there is insufficient separation of the disks, heating and even ignition of the oil sometimes result.

4 Clutches wear most rapidly when allowed to act as cut-off couplings. The disks wear unevenly when mounted on a shaft of small diameter. Worn out disks generally appear like Fig. 3. Upsetting the keyway in disks was frequent.

5 The less the difference between the inside and outside diameters of the disks the more nearly parallel the surfaces after they become

worn, and the more efficient the surfaces; assuming, of course, the area of the disks to be sufficient to meet the requirements.

6 The following rules were finally adopted in designing:

a Select materials with as great a coefficient of friction as possible. (Cast iron and elm or white wood shoes were used.)

b The mean radius of the disks should be as large as possible.

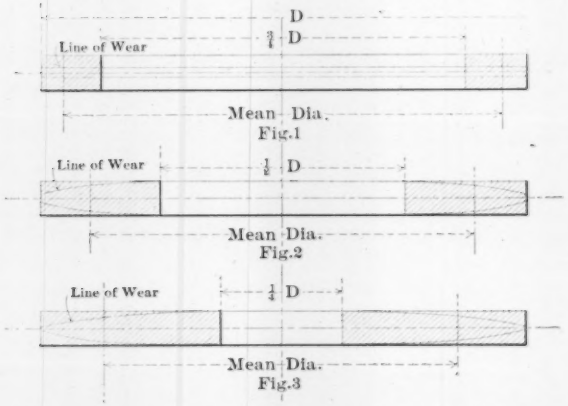
c Reduce the number of surfaces to the minimum.

d Use graphite lubrication.

e Balance the pressure due to gripping and avoid end thrust.

f Have plain lever application with two angles on sliding collar, one to grip and one to free after release.

g Insist on good workmanship, hardened points and true surfaces.



DIAGRAMS ILLUSTRATING WEAR OF CLUTCH DISCS OF DIFFERENT MEAN DIAMETERS

7 A clutch designed along these lines was patented by Thomas Weston, 1899 or 1890, assigned to the Yale and Towne Manufacturing Company and later sold by them to the Brown Hoisting and Conveying Company, Cleveland.

8 The following data may be helpful: Materials for friction surfaces, iron and white wood; minimum pressure, 100 lb. per square inch; maximum pressure, 200 lb. per square inch; coefficient of friction, 0.07 on each sliding surface; hand pressure on lever, 25 to 30 lb.; actual pressure on friction surfaces, 5500 to 11 000 lb.; incline on sleeve, 7 to 22; total travel of hand lever, 22 in.; ratio of hand

movement to shoe movement at the time pressure is applied, 215 to 1.

9 Fig. 1 shows the wear to be practically uniform with hole $\frac{3}{4}$ D. Fig. 2 shows the approximate curve of worn surface with hole $\frac{1}{2}$ D. Fig. 3 shows wear with hole $\frac{1}{4}$ D. In the case of Fig. 3 the edge has worn to a thickness of only 0.002. This wear was due to the sliding when the clutch was applied gradually and the drag between the disks when not driving. Keyways and other details are purposely omitted from the sketches.

10 The foregoing results were obtained under the general direction of Mr. John W. Meserve, Mr. Warren Morse and the writer.

Mr. Hiram Percy Maxim Mr. Souther points out the confusing effect of variable lubrication between the engaging surfaces. Fully as much confusion may be caused however by difference in surface velocity between the engaging surfaces, and especially when the surfaces are metal in both cases. Interesting examples of this may be cited from the writer's experience.

2 In a certain automobile clutch it was necessary to transmit a force of 70 lb. at 12-in. radius. The clutch was of the expanding ring, metal-to-metal type. The internal diameter of the drum was 9.5 in. and the metal cast iron. The expanding band was a bronze casting, $\frac{3}{16}$ in. thick and 2 in. wide. It was anchored at one end and actuated at the other. The band was connected to the engine shaft and was consequently the driver. The drum was connected to the car, and consequently was the driven member.

3 This clutch, when run in ordinary light machine oil, and at a surface velocity of approximately 1000 ft. per minute, would at once pick up the stationary or driven member. If snapped in as suddenly as one could shift it, it would pick up its load very gently, and yet inside of three or four seconds time. The coefficient of friction seemed to increase inversely to the surface velocity difference. It started with the gentleness of a boat without at first materially affecting the engine speed, and then, as the speed of the driven member increased it would gently seize, bringing down the engine speed quickly, and take full advantage of the fly-wheel inertia.

4 But it so happened that 1000 ft. per min. surface speed was lower than what was practically necessary in service, in order to prevent stalling the engine when starting the automobile. The engine speed had to be about 800 r.p.m. to insure starting the car at all times, and this meant approximately 2000 ft. per min. surface speed

on the part of the expanding band in the clutch. At this velocity the characteristics of this clutch were absolutely different. Where the load of 70 lb. could be picked up nicely in three or four seconds time before, it could scarce be picked up at all at this higher speed. The coefficient of friction between the surfaces at the higher speed was but a small per cent of what it was at the lower speed.

5 In my struggles to get out of this difficulty an interesting example of the confusing effects of variable lubrication, of which Mr. Souther speaks, was presented.

6 As explained, at a surface velocity difference of 2000 ft. per min., light machine oil would prevent the metal surfaces from getting together, and this in spite of any combination of oil grooves I could arrange. The question then arose as to whether there was any other lubricant which would not tend so strongly to keep the metal surfaces separated. I thought at once of graphite as a non-liquid lubricant and one which would therefore not flow. I tried it and found to my astonishment that there was no practical velocity difference at which I could not pick up the load as quickly as I wanted. Indeed, with excess of graphite the clutch would go to the opposite extreme and be savage. The difficulty was overcome and the solution is an interesting instance of what Mr. Souther has pointed out regarding lubrication between the clutching surfaces.

7 In leather-to-metal clutches the characteristics of engagement are entirely different from what they are in metal-to-metal. Difference in surface velocities has not as much effect. I have always considered the reason for this was the fact that the leather being porous, any lubrication on the surfaces has an opportunity actually to sink into the fibre of the leather and escape.

8 The approximate uniformity of friction at all velocities in the case of leather is a disadvantageous feature in some automobile services. To insure against slipping, pressures have to be used which make savage starting very difficult to avoid. In racing this is especially bad, for powers are so high as to produce terrific strains. There have been many cases of rupture of gears and shafts in the transmission line of automobiles due solely to the savage engagement of the clutch. The writer has raced many different types of automobiles fitted with both leather faced and metal faced clutches, and the comparison of the two types is instructive.

9 In track and beach racing, the contestants are either lined up directly on the starting line and sent off at the crack of a pistol shot, or they assemble 100 yd. back of the start and line up as they bear

down to the line. In either case the man with the best clutch has an immense advantage. The get away is of course the all important thing, as the one able to sweep ahead in the first few yards is able to get the pole, and on a circular track this advantage is the equivalent of several horse power.

10 With graphite in a metal-to-metal clutch I have won several races over cars able to beat me badly on a dead straightaway. In all races where I used this type of clutch my practice was to begin speeding up the engine as the starter began counting the last three or four seconds. By the time the pistol shot came the engine would be running very likely as high as 2000 r.p.m. At the crack of the pistol I would drop the clutch in with a snap—actually letting it all the way in as quickly as I could release my foot. Without the slightest jerk, it would gather headway very much as a modern high speed elevator does. It would not waste any energy in slipping the driving wheels, nor would it waste a material amount of the inertia of the rapidly spinning fly-wheel. While I raced with this clutch I do not remember a case where I was not able to take the pole from my adversaries in a very few car lengths. I personally do not believe a leather faced clutch could be made which would accomplish this.

11 The metal-to-metal clutch is slow, however, in gaining popularity. This is, in the opinion of the writer, due to its higher cost and greater manufacturing difficulties. In the simple cone type of leather faced clutch the cheapest construction is possible and in many cases a good enough action is secured. In the writer's opinion however, it can never compare with the metal-to-metal type for truly good clutch performance.

Mr. Charles Wallace Hunt A long experience in the building of several varieties of friction clutches prompts me to add to Mr. Souther's paper some data relating to the coefficient of friction of the sliding surfaces of friction clutches, and also to discuss some paragraphs in order that designers may not be misled by hasty inferences drawn from the statements therein contained.

2 A friction clutch of the type under discussion is used to connect shafts in motion, in order that the shock caused by using inelastic jaw clutches may be avoided. When the two parts of a clutch are engaged, the friction surfaces slip, and in consequence are heated in proportion to the mechanical work expended in the slipping of the friction surfaces. When shafts are connected once or twice a day, for example a line shaft in a factory, the heating effect is soon dissi-

pated. When clutches are frequently used, as in the case of countershafts of lathes, the heating effect may be evident, but the rise in temperature is so slight that the heat is dissipated before the next engagement of the clutch. When the friction clutch is used in such work as hoisting coal, where loads of from one to several tons must be hoisted two or three times a minute for hours in succession, the dissipation of the heat is the governing factor in the design.

3 It is necessary for the designer to know in advance whether the revolving shafts are to be disengaged while the power is still being applied to the driving shaft, or at a time when no power is being applied to the driving shaft. This condition is an important factor with any type of clutch, but especially is it so with the cone type. It is the governing one in deciding upon the mechanism for operating the moving parts, and it also limits the cone angles of the friction surfaces. It is the question which must first be decided, and its neglect has been the cause of serious disappointment in many cases.

4 The centrifugal force, which in a rapidly revolving clutch tends to throw the lubricant from between the friction surfaces, is also of great importance in some designs.

5 The statement in Par. 53 carries the idea that theory does not enter into the design of friction clutches, which is misleading. It certainly can and should enter into the design, and that various engineers have not used it is only evidence that they have not applied the knowledge that is available in this class of mechanism. The coefficient of friction in service is not unknown, but it is as definite as the strength of a steel bar, or the physical characteristics of any of the ordinary engineering materials.

6 There are but few materials now used for making the friction surfaces. These are leather, wood, cork and metal. While other materials have been used, they vary but little in type from one of these materials and can be safely classified with one or the other of them.

7 For the design of commercial machinery, if the designer uses a coefficient of friction on iron, of 0.35 for cork, 0.3 for leather, and 0.2 for wood, he will not be disappointed in the result. These figures are derived from a long and varied experience, and are to be used for clutches where the sliding surfaces are kept slightly oiled, a condition easily maintained in ordinary service.

8 When these friction clutches are liable to be flooded with oil, the clutches will work well if an adequate provision for disposing of the surplus oil is provided. The following example illustrates one

method. A clutch similar in type to that shown in Fig. 13 and 14, but with metal to metal friction surfaces, and a different mechanism for expanding the inner ring, has long been used. They have been in use without change of proportions for about 15 years, being thrown in and out of engagement from four to five hundred times per day. The sliding surfaces were about 14 in. in diameter, with a working face of $2\frac{1}{2}$ in. With a stress on the friction surfaces of from 1000 to 2000 lb. the clutch was adjusted to slip from $\frac{1}{2}$ to $1\frac{1}{2}$ in. when thrown into engagement. The expanding mechanism was correctly adjusted to secure this result under the ordinary conditions of service, but when flooded with oil, as it was liable to be, the parts would slip around and make one-half to a whole revolution before the surfaces would engage. A groove was then cut across the friction surfaces longitudinal with the shaft, $\frac{1}{8}$ in. wide and $\frac{1}{16}$ in. deep. This cross groove let the surplus lubricant out, and shortened the slip of the clutch. Additional grooves were cut across, until the slipping, when flooded, did not exceed the $1\frac{1}{2}$ in. which had been fixed as the limit of movement. The distance apart of the grooves to accomplish this purpose proved to be a little over one inch, or a little less than the distance it was desired that the surfaces should slip.

9 Cone clutches have long been built in quantities for hoisting coal, pile driving and similar classes of work. Those built by leading companies in New York and Newark usually have the cone surfaces at an angle of from 20 to 22 deg. between the axis and the conical surface. These clutches require a pressure on them constantly while at work, but free themselves promptly either standing or running. The cone frictions built by another company have an angle of from 18 to 20 deg., but are always accompanied by an operating device with which to pull the friction out as well as to press it in. The axial motion of a cone clutch of 20 deg. 36 in. in diameter, in regular service is from $\frac{1}{2}\frac{1}{0}$ to $\frac{1}{16}$ in. The loads hoisted by this clutch gave about 6000 lb. pull on the friction surfaces, which were leather and cast iron. They are engaged and disengaged four or five hundred times per day. The friction leather lasts for several years.

10 Contrary to the statement in Par. 70, the engagement of a cone clutch is not a difficult problem, as large numbers of these are in daily service, operating without the slightest difficulty. The method of moving shown in cut 18 is admirable, and there are many other devices, the essential feature of which is the absence of lost motion, and great rigidity in the operating mechanism.

11 Cone clutches used on automobiles have usually been designed with a small angle, that is from 7 to 13 deg., and with leather friction surfaces. These small angle clutches will not disengage when standing still, or when the power is not driving the machine.

12 The mechanism in current use on automobiles for withdrawing these clutches almost invariably has considerable spring, and sometimes slack motion in the joints in addition. This construction makes the cone clutch irregular in its action, and accounts for the complaints of the "fierce" action both in engaging and disengaging. If a rigid handling mechanism is used, and a larger angle cone, the difficulties will all disappear. The cone clutch then becomes admirably smooth in its engagement and disengagement. The moving mechanism shown in Fig. 18 is especially rigid in construction and satisfactory in automobile service.

13 The flat surface clutches used in automobiles manufactured by the De Dion Company of France, Fig. 35, and the American type made in Detroit, Fig. 18, may be considered mechanically as cone frictions of 90 deg. They work with admirable smoothness, and would also work equally well with a conical clutch if the cone angle was 20 deg. or over. The strength of the springs needed to hold the surfaces in contact increases as the cone angle increases. The cut in Fig. 18 shows a clutch that is normally off, but the same makers also build for large cars a double-faced clutch of the same type, which is normally on.

14 Leather is usually used on the inner, and iron on the outer cone. It may be interesting to note that chrome leather wears equally as well as ordinary bark-tanned leather, and is very much superior when the surfaces run hot. When the clutch is large in diameter and the amount of slipping is great enough to heat the parts, then a difficulty sometimes arises from the fact that the leather is a non-conductor of heat, and when heat is generated at the point of sliding contact, the metal of the outer shell of the clutch becomes hotter, and consequently increases more in diameter than the inner one which supports the leather. If when a clutch is so heated it is left engaged until it cools, the outer cone will shrink more than the inner one, binding the leather so tightly that it is difficult to disengage the two parts of the clutch. For this reason, large size clutches are sometimes made with the non-conducting material attached to the exterior part of the cone, so that the inner cone will be the hotter one, then when the clutch cools, the inner cone will shrink away from the outer one and thus loosen the clutch.

15 Par. 76, 77, 78 and 79, and cut 20 show a form of construction that is short lived. The leather over the springs that first comes into contact will promptly wear through, and the clutch will soon be ruined by the breaking of the leather. A larger angle of cone surfaces would at once solve the problem.

16 Par. 88, 89, 90 and 92 describe clutches that are normally held in driving contact by a spring. This is the most suitable clutch for an automobile. Various operating devices may be used, but nearly all of those shown are faulty in one respect, that is there is a springy or a loose-jointed method of moving the cone in and out of contact. To avoid irregular working of the clutch, the mechanism must be rigid and the joints accurately made.

17 The impossibility of rapidly dissipating the heat generated when heavy and rapid work is required eliminates from consideration enclosed clutches such as shown in cuts No. 8, 38, 39, 40, 41 and 42. They are frequently satisfactory on automobiles, as the service is not continuous. The oiling in this type requires care in the design. Frequently it is inadequate, or too intermittent to keep the surfaces in an equable working condition.

MR. FRED MILLER Clutches now in existence operate on the principle that one clutch cannot be thrown into engagement until another is thrown out, and during the interval between these two operations no power can be transmitted. We have the best illustration of this in the transmission gear of an automobile. In order to change from one speed to another, you must disengage a clutch, and thus engage a clutch which will drive at the new speed. Between these two operations there is a time interval during which no power is transmitted.

2 To overcome such intermittent action I have designed a clutch which continues to drive until the load is taken away from it by the engagement of a second clutch. For instance, suppose one clutch to be engaged and the train of mechanism to operate at the resulting speed. To increase that speed, simply throw in another clutch which will take the load away from the first one and drive the train at a higher speed without interruption to the driving action. Assuming that the train is now operating at this higher speed and it is desired to go to the lower, disengage the clutch which is driving at the higher speed and the train will fall off in speed until it reaches that of the lower speed clutch, when the power will be transmitted by the lower speed clutch.

3 The principle which permits the higher speed clutch to take

the transmission away from the lower, and vice versa, is that these clutches grip only when rotation is in one direction and rotation in the opposite direction immediately relieves the grip. When we engage a higher speed clutch, it is equivalent to rotating the lower speed clutch in the opposite direction, and the load is immediately taken away from that clutch and assumed by the higher speed clutch. The same principle holds true when we disengage the higher speed clutch. Rotating that clutch in the opposite direction the speed drops until the speed at which the lower clutch grips is reached, when the lower clutch assumes the transmission.

MR. E. J. MCCLELLAN The practice of The Garvin Machine Company is to use cone frictions on automatic tapping machines, with an angle of 10 deg. to the center line where clutch is thrown in by toggle, and 6 deg. where thrown in by a comparatively weak spring. There is little wear on the friction surfaces if clutch is thrown in sharply. The only wear comes on the shoulder that presses the clutch in. On countershafts and friction head screw machines we use expansion rings. In this style of clutch centrifugal effect can be neutralized by turning up the clutch ring when in the expanded position so that there is an initial contracting stress in the ring. By this means we have no trouble up to some 700 r.p.m. A countershaft clutch ring of 9½ in. diameter and 1½ in. wide, expanded by a toggle, will resist a pull of 800 lb. on its rim.

2 One peculiarity of the Weston clutch is the accuracy with which it can be manipulated. We use a small one to control the reversing mechanism of a table on a slot milling attachment. This little clutch responds so accurately to the action of the tripping dogs, that the travel of the slide does not vary two-thousandths of an inch per stroke.

MR. J. J. BELLMAN The very complete paper by Mr. Souther on the subject of clutches shows that he has given the matter extended study. The work however which has come directly under his notice appears to have been in the form of clutches adapted for light high speed power transmission as found in automobile work. Clutches however are used for many forms of heavy transmission work, for which the forms most fully described are not suited.

2 The clutches best known for the heavy class of transmission such as the winding drums of hoisting engines, or the operating mechanism of dipper dredges, are usually of the cone, band or umbrella type. With the cone clutch the practice is to force the parts together with a considerable pressure and take up the resulting end thrust by means of collars.

As is well known any end thrust is a serious objection, not only in frictional losses but in the means necessary to absorb it. With the cone type, the question of adjustment is a very serious one as is frequently pointed out in Mr. Souther's paper. With heavy shafting, it is impossible to adopt the self-centering shaft which he considers almost essential for the proper operation of the clutch. Therefore, efforts have been made to use forms of band brakes which are susceptible of easier adjustment. Much mechanical ingenuity has been applied to the means for applying the band between the two revolving elements, the most successful being by means of a steam cylinder operating on a wedge which in turn operates on links and applies the band clutch, or a hydraulic cylinder on the driver connected directly to the band and

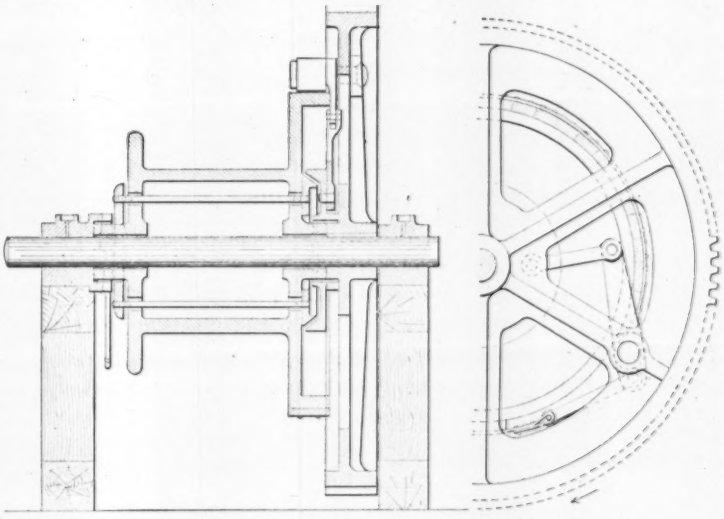


FIG. 1 FIG. 2

MEANS FOR OPERATING CLUTCHES

actuated by water supplied through the centre of the shaft. That these may rotate with shafts, one of the essentials for this form of applying the band, the means for applying the mechanism must be placed at the center by boring the shaft. This boring necessitates increased weight for the shaft and increased expense for the operating mechanism.

3 A method of applying the band clutch so simple as to be of interest to those dealing with the heavier kinds of clutches in particular, has recently been devised by Mr. Gilbert H. Gilbert. The drawing shows the ordinary form of hoisting drum where the main gear is operated continuously by a constant speed engine, and the hoisting

drum is started and stopped at the will of the operator; the drum is connected to the main gear by means of an ordinary form of band brake. To tighten this brake, an auxiliary cone friction is used, one part free to revolve on the shaft, and the other attached to the hoisting drum. Pressure applied to the part attached to the hoisting drum makes it engage with the part free on the shaft, and tends to retard its revolution. A link is so connected that by the engagement of the part attached to the drum with the part free on the shaft, the band brake is applied and the drum, auxiliary clutch and gear revolve together.

4 The following are some advantages claimed for this form of construction:

a The end thrust for cone frictions, usually figured at from 15 to 20 per cent of the total power expended, is eliminated;

b The distance between bearings can be reduced materially over any other form of clutch. This will allow the use of smaller shafts for the same strength;

c Shafts can be made solid, eliminating the necessity of boring same to operate the ordinary steam or hydraulic form of band friction;

d A number of drums may be placed on the same shaft, as it is not necessary to expose the end of the shaft for the purpose of installing steam or hydraulic cylinders;

e Hydraulic or steam cylinders entirely eliminated, with their attendant expense for first cost and operation;

f Adjustment of band frictions is much less frequent than the adjustment of cone frictions. The reliability of a band friction is much greater than that of a cone friction;

g The speed of operation is greater, as only one lever is necessary to operate this clutch;

h Hoisting and lowering may be done on the one brake;

i The size of the whole equipment of a heavy machine can be reduced both in floor space by the elimination of the cone frictions or clutch operating cylinders, and also by the reduction in the size of the boiler and hoisting engine. This latter advantage is due to the fact that the end thrust usually consumes from 15 to 20 per cent of the power of the hoisting engine.

j The cost of the construction submitted is considerably less than that of any other form of friction for heavy machines.

k Cost of operation is reduced to a minimum as the power to operate the clutch is received from the main engine, and losses due to steam or hydraulic operating cylinder for the friction are entirely eliminated.

MR. R. M. GAY In his reference to the Sturtevant clutch, Mr. Souther mentions that as the clutch depends for its action upon centrifugal force, one could not, in the case of an automobile, speed up the engine, throw in the clutch, and jump the machine out of a mud hole. While this is true, the clutch has a peculiar action when driven by a gas engine, which makes it effective at low speeds.

2 At the instant of explosion in a gas engine the motion of the flywheel is greatly accelerated. This causes the weights of the clutch to grip harder at that instant, giving the clutch excessive torque. When you get in a mud hole and the speed of the engine drops, at each explosion the flywheel accelerates, the torque increases beyond the normal torque of the engine for the moment, and the clutch takes hold. After a fraction of a second, if the engine has not been able to pull out, the speed of the flywheel and the torque will drop, the clutch will let go and the engine will run freely, accelerating again under the next explosion. Thus the engine does not stall, but continues to pull all it can.

3 Many times I have been able to pull up grades with this clutch when the engine was running at comparatively slow speed, where with an ordinary clutch it could not be done. The effect is the same as though you threw in the clutch for a fraction of a second when the engine was exerting its extreme force and before you stopped the engine pulled out the clutch and allowed the engine to accelerate again. The centrifugal clutch does this automatically; and the engine never can be stalled.

ELMER H. NEFF As some friction pulleys as applied to machine tools may be of interest in this general discussion, I desire to present four styles, made and used for some years by the Brown & Sharpe Manufacturing Company. Two of these pulleys are for countershafts and are self-oiling. The other two are used on spindles of screw machines. I think these may be of assistance to designers in general, as showing the proper relation between the friction surfaces and the belt surfaces to insure a suitable balance between driving capacity, speed and durability. The friction surfaces of all these pulleys are metal to metal.

2 The friction pulley shown in Fig. 1 will be readily understood from the cut. These are referred to in Table 1 as old designs, and are for counter-shafts of milling machines and other machines where the speeds are moderate, and the necessity of reversal or stoppage is comparatively infrequent. There is a cast iron pulley *A* against

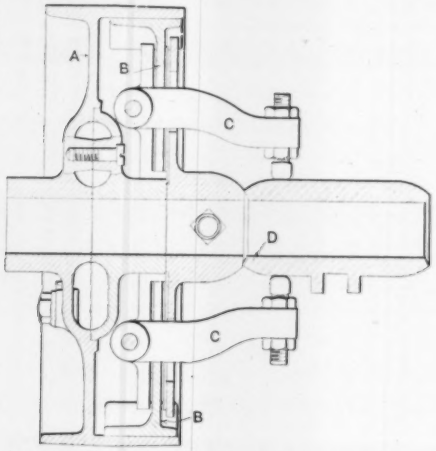


FIG. 1 FRICTION PULLEY GENERAL DRAWING

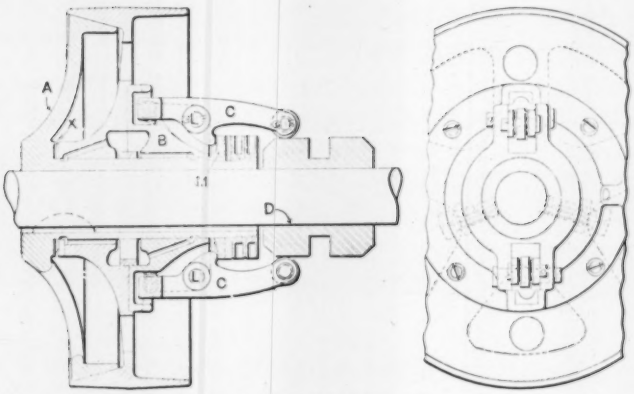


FIG. 2 FRICTION PULLEY

the inner surfaces of which are brought two friction segments *BB* through the action of the levers *CC* and thimble *D*. When the thimble *D* is slid under the ends of the levers *CC* these segments are pushed apart radially until they engage the pulley. The limiting speeds which are recommended are given in Table 1, and are approxi-

mately 900 ft. per minute at the friction surfaces. Higher speeds would make it difficult for these pulleys to release on account of the action of centrifugal force on the segments *BB*. Long experience has shown that these speeds are about correct for satisfactory continuous service.

3 The friction pulley shown in Fig. 2 is the newest design for counter-shaft purposes, and is used for screw machines and in other places requiring high speed with frequent starts and stops or reversals. It consists of a belt pulley with conical friction, having the friction surfaces at an angle of 15 deg. At the limiting speeds given in Table 1, the speed at the medium diameter is approximately 1200 ft. per minute. The operation of this pulley is, that by sliding the thimble *D* under the rolls in the two levers *CC* which are pivoted on the friction body *B* by pins *LL*, pressure is exerted at the heel against the collar *M*, drawing the friction cone *A* into the pulley. A flat spring between the hub of the pulley at *X* and the friction *A* serves to separate them when the levers are released. There is a chamber for oil, and a ring oiler with passages as shown, so that the oil circulates between the bearing surfaces, and back to the chamber again. The friction *A* is fastened to the shaft by set screws and also by a Woodruff key. Further details regarding this pulley are given in Table 2.

TABLE 1 OLD DESIGN FRICTION PULLEYS

Diameter in.	Belt in.	Highest speeds	H.P. PER 100 REV.		Diam. of friction surface in.	Width of friction surface in.	Chordal length of each shoe in.
			Single belt	Double belt			
8	2¼	450	0.59	0.93	7½	1⅜	3½
10	3	375	1.00	1.41	9½	1 1/16	4 7/16
12	3½	325	1.37	1.96	11 47/64	2 5/16	5¼
14	3½	275	1.58	2.30	13 43/64	2 1/32	6 5/16
16	4	250	2.36	2.92	15¼	2 13/16	8
18	4½	225	2.65	3.78	17¼	3 5/16	8

TABLE 2 NEW DESIGN FRICTION PULLEYS

Diameter in.	Belt in.	Highest speeds	H. P. PER 100 REV.		Largest diameter of friction in.	Width of friction surface in.
			Single belt	Double belt		
8	2½	500	0.72	1.03	7⅞	1 1/16
10	3	500	1.10	1.56	9⅞	1 5/16
12	3½	450	1.52	2.18	11⅞	1½
14	4	350	2.00	2.90	13⅞	1 11/16
16	4½	325	2.62	3.70	15½	1¾

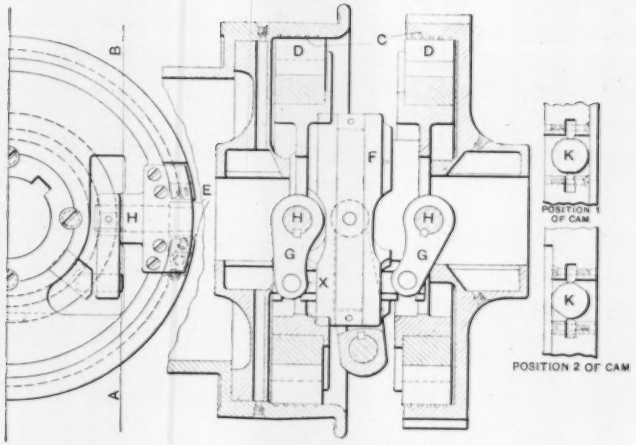


FIG. 3 SECTION OF THE CLUTCH MECHANISM

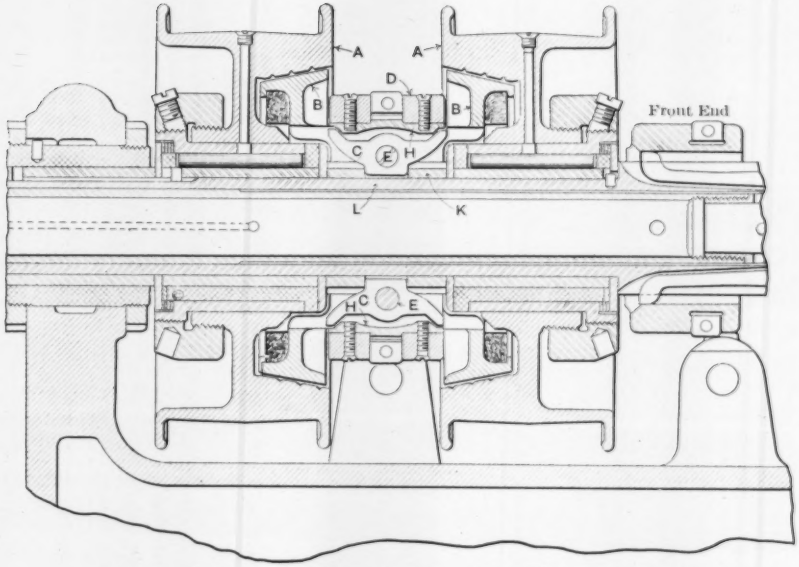


FIG. 4 SECTION THROUGH SPINDLE OF AUTO-SCREW MACHINE

4 In Fig. 3 is illustrated a friction clutch for use on comparatively slow speed screw machine spindles or similar work. This consists of friction surfaces on the inside of the gear or pulley to be clutched as shown in the figure at *C*. Inside this is the ring *D* with a portion cut out at point *E*. By the operation of a system of levers slider *F* is thrown in the direction desired. This slider bears on the levers *GG* at *X*, which rotates the shafts *HH*, on the ends of which are the cams *KK*, as shown in Fig. 3. The rotation of these cams referred to, which are in the space cut out on one side of the friction rings, expands the ring into the friction pulley.

5 The system of friction pulleys for the spindles of automatic screw machines is illustrated in Fig. 4, Table 3 giving the data regarding speed and size of these pulleys for these machines. The pulleys are cast iron and the friction bodies *B* of bronze. The friction surfaces are at an angle of 12 deg. and metal to metal.

TABLE 3

Automatic screw machines	Largest diameter	Belt	Width of friction surfaces	Maximum speeds
No. 00	$3\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	2400
No. 0	$4\frac{3}{8}$	2	$1\frac{1}{8}$	1800
No. 2	$5\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{1}{4}$	1200

6 The operation of this clutch is, that by means of a system of levers the ring sleeve *D* in which are two tool steel shoes *H*, is moved either to the right or left over the ends of the dogs *CC*, which are pivoted on the pins *EE*. The heels of the dogs *CC* work in slots in the sleeve *K* on the spindle *L*. When the sleeve *D* moves over the dog *C*, its action is to rotate it on the pin *E*, but as the heel of the dog *C* is against sleeve *K*, and that is immovable endwise, being rigidly attached to the spindle, the whole friction body *B* is compelled to move endwise and is thus brought into engagement with the pulley *A*. The design of the mechanism is such that, by means of cams, this friction is thrown into engagement with the pulleys instantaneously. One friction pulley is running in one direction and the other in the opposite direction. In performing a threading operation, as the die approaches the end of its cut, the lever is tripped, the sleeve *D* is thrown in the proper direction, engaging the opposite clutch, and the spindle is therefore instantly reversed.

7 In the smallest size machine, the No. 00, with a maximum

speed of 2400 r.p.m. we have a mean speed of about 2200 ft. per minute. As the pulleys are running in opposite directions, in order to accomplish the reverse, the friction body *B* must be thrown out of engagement with the pulley at the left, which is running at a speed of 2200 ft. per minute, into engagement with the pulley at the right, which is running in the opposite direction at a speed of 2200 ft. per minute, making the relative speed about 4400 ft. per minute at the maximum. These machines have run at these speeds for years without the necessity of replacement, showing that the friction takes hold of the pulley with practically no slip, otherwise there would be wear. It is possible to thread with these up to a shoulder, using a solid die holder with no side play whatever, and it reverses without damage to the thread being cut, or to the die holder, or to the machine.

ECONOMY TESTS OF HIGH SPEED ENGINES

By Messrs. Dean and Wood, Published in June Proceedings

Mr. G. A. Young I value the results of the tests under discussion for two reasons: first, because the opportunity to secure reliable data from different types of the same class of engines is rare; and second, because all the engines here reported were working under their normal conditions. The steam consumption is well within the range experience has taught us to expect from high speed engines. I was surprised to note, however, that the engine with the one flat slide valve surpassed in economy the four-valve type of engine.

2 Thinking that others might be interested, I have compiled a table of comparative results of tests made by me in recent years on six high speed engines. A brief description of each of the engines follows:

Engine No. 1: The engine reported on this test was run about five days after installation, before the test was made. It was directly connected to a two stage centrifugal pump and had a piston valve.

Engine No 2: This engine was five years old and used a plain, unbalanced D valve. It was directly connected to a large fan.

Engine No. 3: The valve on this engine is similar to that on Engine No. 1 described by Messrs. Dean and Wood. It has seen 15 years of service.

Engine No. 4: Engine No. 4 was directly connected to a 75 kw. generator. It used one flat valve, working between the seat and a pressure plate, self-adjusting for wear. The engine had been run for two months when the test was made.

Engine No. 5: This engine was fitted with four valves of the rotary type, operated by two eccentrics. The test given was the acceptance test after the engine had been in service about eight months furnishing power for a lumber mill.

Engine No. 6: The valve on this engine is of the piston type, with a riding cut off valve. The valve chest and cylinder had been renewed five years previous to the test.

All the engines except No. 2 had shaft governors.

TABLE 1

1	Number of engine	1	2	3	4	5	6
2	Horizontal or vertical	V	H	H	H	H	H
3	Load on engine	centrifugal pump	fan	brake	generator	lumber mill	brake
4	Time of service	5 days	5 yrs.	10 yrs.	2 mos.	8 mos.	5 yrs.
5	Dimensions of cylinder, in	10 by 12	8 by 12	7 by 12	13 by 12	16 by 22	7¾ by 15
6	R.p.m	325.5	250	225	268.6	200	220
7	Steam pres. lb. gage	146.4	140	125	120.3	79.6	140
8	Quality of steam, per cent	98.4	98.3	99	98.7	97.3	98
9	Back pres. lb. gage	1	2	2	12	4.8	1
10	I.h.p	69.4	30	28	130	123	45
11	Lb. of steam per i.h.p.h	46	38	36	34.6	31.7	30.4

3 All the engines were working at about their normal rating, so that the results given in the above table are as good as could be expected. Engine No. 6 shows the best results, Engine No. 5, with the four valves, coming next.

4 When comparing the steam consumpt on of these engines with those given in the table accompanying Par. 46 of Messrs. Dean and Wood's report, it must be remembered that their results have been reduced to atmospheric conditions, and also that the horse power of the engines in my table is generally lower.

PROF. THOMAS GRAY In connection with the comparative economy of small sized turbines and slide valve engines, it may be interesting to state the results of a recent test on a 100 kw. turbine. This turbine was of the Curtis type, and direct connected to a 100 kw. direct current dynamo. The results showed a consumption of about 60 lb. of steam per kilowatt hour, delivered to the working circuit.

2 Making allowance for the dynamo efficiency, this result compares favorably with those quoted for slide valve engines.

MR. C. H. TREAT Comparison of water rates for quite small engines with those given may be of interest. This contribution gives water rates for an American Blower Company 7 in. by 7 in. engine, non-

condensing, after hard service and badly worn, and also rates for engines similar to this when in proper condition.

2 The engine had been used to drive a fan for forced draft, and was displaced by larger equipment. The automatic governor (Rites) was put on the engine for the test and the valve was set for equal cards. The valve (Fig. 1) was 0.003½ in. smaller than the bore.

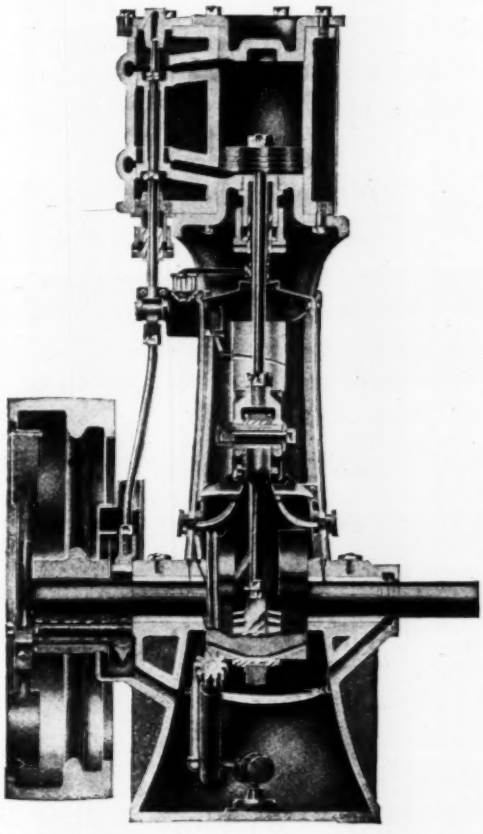


FIG. 1 SECTION SHOWING ARRANGEMENT OF CYLINDER AND VALVE CHEST

3 The cylinder had been cut very badly, or from 7 in. bore to an average diameter of 7.082 in. and was quite uneven. The piston was also worn small (7 in. – 0.013 in.) and the rings were broken. New rings were put in the piston.

4 The test was made by condensing the exhaust in a barrel of water on sensitive scales. Time after exhaust pipe dropped below surface of water until it was pulled out was taken by two good observers whose results agree closely. The temperature of water before and after exhausting into it was taken and affords a check. Quality of steam to the engine was taken by a throttling calorimeter. Indicator cards were taken in rapid succession while the exhaust was going into the barrel.

5 A small correction for water spilled out when the exhaust pipe was drawn up was added. It was determined by the time the exhaust could be into the barrel of water without changing its weight; that is, with engine loaded for test No. 1 it was found necessary to exhaust into the barrel of water practically two seconds to condense enough to make up for the spill when pipe was withdrawn. From the test $\frac{1234 \text{ lb.}}{60 \times 60} = 0.34$ lb. condensed per second, and 0.7 lb. was added as the correction. In every case engine was run until conditions were constant before starting test.

6 When it is considered how loose the piston was, and how uneven the cylinder, the effectiveness of piston rings in preventing leakage is quite marked and emphasizes the importance of keeping them up.

	Test No. 1	Test No. 2	Test No. 3
Date	July 10, 1908.		
Previous length of service	10,000 hrs.		
Size of cylinder	7.082 in. mean diam. by 7 in.		
Diam. of piston rod	$1\frac{5}{16}$ in.		
Engine belted to	Centrifugal fan wheel 30 in. diam.		
Approximate load	Full	1.2	Half
Quality of steam	98.7	99.2	99.2
Average steam pressure, pounds	93	96	90
Average revolutions per minute	430.5	432	435
Indicator Springs	80	80	80
Indicated horse power	30.3	36.6	17.4
Indicated friction horse power	1.18		
Indicated friction per cent	4		
Water condensed per hour, pounds	1234	1535	808
Water per indicated horse power per hour observed, pounds	41	42	46.3

7 The data for full line curves, Fig. 2, showing results for engines in proper condition, were obtained from tests made at the American Blower Company Plant in 1905. Test was made by condensing the exhaust in a surface condenser consisting of a steam heater section

pipe immersed in a tank, having an overflow and one pound supply from city water. The whole was up overhead so that any drip or leaks could be seen and stopped. That the section itself did not leak at any time was shown by the entire absence of drip, unless steam was entering.

8 The condensation being piped down, heater pipes and base were slanted to drain through a water seal to prevent steam blowing

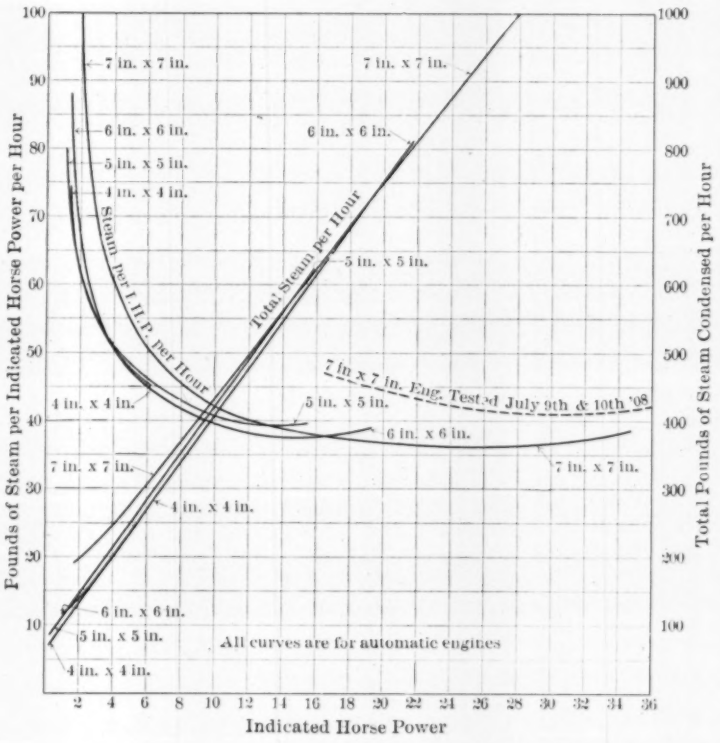

Pounds of Steam per Indicated Horse Power per Hour
Total Pounds of Steam Condensed per Hour
Indicated Horse Power
Steam per I.H.P. per Hour
Total Steam per Hour
7 in. x 7 in.
6 in. x 6 in.
5 in. x 5 in.
4 in. x 4 in.
7 in. x 7 in. Eng. Tested July 9th & 10th '08
All curves are for automatic engines

FIG. 2 PERFORMANCE CURVES OF HIGH SPEED ENGINES

into the upper of two barrels having drain valves at bottom, the lower barrel being on platform scales. For each barrel of water the weight before filling was subtracted from the full weight. The scales used weighed correctly. The indicator spring was calibrated by both the makers and ourselves. A long lever reducing motion was used to prevent lost motion. The drum spring was drawn tight, and at the high speeds cards were short. We indicated at speeds up to

500 r.p.m., using a Prony brake at high speeds and taking the brake horse power.

9 Readings were taken every five minutes after conditions had become constant and the engine was well warmed up. No corrections were made for moisture in steam but a few determinations show a quality of 98 to 99 per cent. Data given in Fig. 2 are from tests of a number of engines for most sizes, the speed being 350 to 500 r.p.m., though some were at somewhat lower pressure.

AIR LEAKAGE IN STEAM CONDENSERS

By Thomas McBride, Published in June Proceedings

Mr. Charles A. Howard The author has taken a step in the right direction in this paper, as very little attention has been paid to the question of air in surface condensers, and its quantitative measurement.

2 I desire to discuss the statement of the author to the effect that the purchaser of a condensing equipment should be required to specify quantitatively the amount of air to be handled by the dry air pump. He says, "It is only as fair to expect that purchaser's specifications shall at least limit the amount of air to be handled by the air pump, that the manufacturer may intelligently select the size of this pump, as it is to expect these specifications to meet the amount of steam to be condensed, that the manufacturer may properly determine the size of the condenser." I think there is very little justification for this point of view, which seems extremely one-sided. If the author had gone a step further and said that it was as reasonable to expect the purchaser to specify the amount of air to be handled, as for him to specify the number of square feet of surface to be installed, his statement would seem more logical. In other words, if the viewpoint taken contemplates merely the furnishing and erection of the cast iron and bronze which forms the condensing unit without regard to its design or operation, of course the author's statement is correct and fair, but this method of handling the contract,while no doubt very desirable to the manufacturer, would be rather bad business for the purchaser and is adopted only in exceptional cases.

3 When the purchaser buys an engine or a turbine, it is guaranteed to complete a certain performance conditional among other things on the maintenance of a definite vacuum at its exhaust nozzle. I think I am safe in saying that none of the engine or turbine manufacturers would make a guarantee on the minimum amount of air leakage through their machines. Even if this were done, it would be practically impossible to ascertain the relative amounts of air entering the system through the prime mover, the connections and the condenser, or originally present in the steam, so the purchaser would

be unable to hold the manufacturer responsible for the air leakage.

4 Hence if the purchaser specifies the quantity of air to be handled by the pump, this amounts virtually to releasing the manufacturer from his guarantee, as, if the required vacuum were not produced the manufacturer's first claim always would be that there is a leak in the system; and as the place where the air entered could probably not be determined to the satisfaction of both parties, the purchaser would have no hold on the manufacturer of either the prime mover or the condenser.

5 It is the duty of the engineer, in drawing up contracts and specifications, to protect his client against loss or damage in every manner possible, and to this end he should call upon the condenser manufacturer for complete guarantees as to the performance of his unit with the specified intake water temperatures, pounds of steam to be condensed, and vacuum to be maintained. This is from the purchaser's point of view only, but I think it can be shown that it is not unreasonable from the point of view of the manufacturer for him to take the full responsibility for the air to be handled by the air pump.

6 The manufacturer's objection is that he is held responsible for an air leakage over which he has no control and to a certain extent he is right, but it must also be considered that, as stated above, the purchaser has no control over the leakage in the prime mover at least. Now why should the purchaser take this responsibility rather than the manufacturer?

7 The largest part of the leakage in most installations is in the connection piping from the prime mover to the condenser, and the most satisfactory method of handling this work is to make a part of the condenser contract the piping from the exhaust flange of the prime mover to the condenser, allowing the condenser man to make the exhaust connection to the prime mover and holding him responsible for that joint. This method of handling the work places under the condenser manufacturer's control the chief source of leakage and makes him responsible for it.

8 The remaining possible leakage is in the prime mover. In a steam turbine of the Parsons type and with the present construction methods of the water sealed glands, this leakage of air is inappreciable in both the Westinghouse and Fullager machines. This statement is made after close observation of several different machines of both makes. In the Curtis turbine, the leakage is frequently very excessive at the light loads, since under this condition the vacuum penetrates to a point high up in the machine, but as the guarantee of the con-

denser is usually asked for at the full load of the turbine or more, this matter need not worry the condenser designer. At full load, these turbines are nearly as tight as the horizontal machines, and the fact that they will hold over 29 in. of vacuum referred to a 30 in. barometer when running at full load, with a ratio of vacuum pump displacement to the volume of the condensed steam of 18, is a fair indication that this leakage is low.

9 It is the business of the condenser designer to understand all these points of air leakage thoroughly, and as a specialist, he should be able to judge the amount of this leakage better than the purchaser, to whom condenser engineering is only one of the many branches of his business.

10 In almost every case, the manufacturer has previously installed his apparatus in connection with the same make of prime mover that is to be attached to the condenser of the installation in question, and therefore he has some knowledge of what the leakage in the prime mover will be, whereas the purchaser has probably had little if any previous experience with the make of prime mover to be used.

11 Therefore it follows that as the piping between the condenser and the prime mover can be done by the condenser manufacturer, he has control over the tightness of the whole system except the prime mover, and should be held responsible for it, and considering the greater opportunities of the manufacturer to learn what the air leakage is, perhaps the most important part of his business as a specialist, it is certainly not unreasonable for the purchaser to hold him responsible for the tightness of the prime mover, over which the purchaser has no control and practically no definite means of ascertaining the amount of leakage.

MR. C. L. HEISLER From a manufacturer's view, it is hoped that Mr. McBride's paper will bring about discussion sufficient to develop eventually more precise and reasonable methods of defining the responsibilities of the purchaser and manufacturer of condensers.

2 No doubt a hearty welcome will be given to a form of specifications or contract, which will make it unnecessary for the builder to resort to the use of an absolutely meaningless guarantee in order to defend himself against an unreasonable demand, when the purchaser asks the condenser builder to be responsible not only for the workmanship, design, and air tightness of his own apparatus, but also the air tightness of the turbine, or reciprocating engine, and the entire exhaust pipe system connected with the condenser.

3 Since the condenser builder has only a small margin of profit, derived solely from the vacuum pump and condensing chamber, it appears hardly fair to burden him with the expense of finding and correcting air leaks in the entire exhaust system and engines, built and installed by others. Even when leaks are found, it is often impossible to remedy the defect immediately and the purchaser refuses to accept, believing that such small leaks should not reduce the vacuum as alleged, and naturally expecting the condenser builder to prove his allegation by correcting the defect at his own expense.

4 A small defect in a piston rod packing will admit sufficient air to cause a serious dispute, if not actual rejection under an iron-clad guarantee, as frequently asked for by purchasers. On an otherwise air-tight chamber it needs only the equivalent of an orifice of $\frac{1}{16}$ in. diam. to reduce the vacuum $\frac{1}{2}$ in., and $\frac{1}{8}$ in. diameter to reduce it $\frac{7}{8}$ in., with the barometer at 29.2. With a dry vacuum pump having a piston of 20 in. diameter and 12 in. stroke, at 120 strokes per minute, a vacuum was maintained at 28.9 in., that is, within 0.3 in. of the barometer previous to admitting air through orifices of above diameters.

5 It appears extremely difficult to deduce formulae and provide means for actually measuring the quantity of air leaking into an exhaust or vacuum system common to steam engines. Then after knowing the quantity it is equally difficult to find where the air comes from and whose work is responsible for the leakage. Meanwhile the condenser builder is waiting for acceptance and payment whereas the other manufacturers who are equally responsible, and perhaps guilty, have already received their payment, and may be enjoying the unhappy condenser man's predicament.

6 When purchasing a vacuum pump, it seems that the purchaser would get better satisfaction if he did not demand an impracticable degree of rarification, but instead asked for a vacuum, say within 0.3 in. or 0.4 in. of the barometric reading, when the dry vacuum pump is running at the rated speed, and when attached to a small air-tight chamber of sufficient volume to steady the mercury column. Such a guarantee, including good design and workmanship, would be a reasonably accurate measure of what is being paid for, and when installed there would be some reason to think the leakage was not in the air pump. The purchaser should also specify the size or approximate volumetric displacement, as he is in a better position to know the conditions of operation than the manufacturer. This would give more uniformity in bids and save considerable unnecessary delay caused by the quite common practice of merely stating the degree

of rarification, and forgetting to state the size and length of the exhaust mains, type or number of engines, and omitting other important data.

7 It is, however, more difficult for the purchaser to specify details of the condensing chamber. He can only insist that it is tested for tightness after erection. He may ask to be given the principal dimensions, volume and weight, with a set of drawings, from which his engineers can study its design and relative points of merit.

THE AUTHOR Mr. Howard reaches the conclusion in his discussion that the condenser designer, as a specialist, should be able to predict the amount of air to be expected from any particular unit, from his experience with similar units. It has not been possible to make such predictions because of the fact that the amount of air in one unit is often many times that found in a supposedly duplicate unit. An instance of this is cited in the paper. In this case, however, the condensers, although both surfaces were of different types, but the leakier of the two units had a much simpler and therefore probably a much tighter condenser. The writer has made rough estimates of the amount of air present in the exhaust steam in a number of condensers, to determine approximately whether it was more or less than that anticipated by manufacturers' ratings of air pumps, and has found the quantity of air very erratic and apparently more the result of accident than of any particular design of prime mover.

2 Mr. Heisler's discussion presents results of some interesting experiments worthy of application to the conditions existing in condensers. Where the $\frac{1}{16}$ in. hole caused the absolute pressure to rise from 0.3 to 0.8 in. the relative humidity of the air reaching the air pump was probably low, and the air pump may have been handling more air than water vapor. In a condenser the air is always saturated and the air pump, in handling a little air, must also take out a large volume of water vapor. The pressure of 0.8 in. absolute corresponds to 73 deg. fahr.; if there had been present a supply of water at this temperature, the $\frac{1}{16}$ in. hole would have made the result impossible except with an air pump of infinite size. It is also evident that the more efficient the condenser, that is, the closer the temperature of the vacuum is brought to that of the intake water, the greater will be the proportion of water vapor to air in the air pump cylinder and therefore the greater the damage from a small leak.

3 It is gratifying to note that both Mr. Heisler and Mr. Howard agree with the suggestion that quantitative measurement of air in condensers is desirable. The only essential to its use as a feature of condenser contracts and their guarantee seems to be the general adoption of a standard method.

MISCELLANEOUS DISCUSSION

IDENTIFICATION OF POWER-HOUSE PIPING BY COLORS

By William H. Bryan, Published in June Proceedings

The Author It has been a source of gratification to the author to note the general interest in this subject, and the favor with which his suggestions have been received. The additional ideas brought forward in the discussion are helpful. I was particularly pleased to note the work already done on the Great Lakes.

2 It would seem well to emphasize some cardinal principles which should underlie such a code, particularly simplicity, few colors and of a permanent character.

3 As to whether the time is ripe for standardization, it is difficult to say. Such a committee, if appointed, should not hurry its report, but take ample time to study the problem both as to American and foreign practice, inviting suggestions as well as reports of the experience of those who have already made a start along these lines.

POWER PLANT OPERATION ON PRODUCER GAS

By Godfrey M. S. Tait, Published in June Proceedings

The Author Replying to C. W. Lummis, as to the efficiency of a producer plant operating with gas as an endo-thermic agent, compared to one using steam, I have found that in theory the economy shows in favor of the steam-operated producer. However, when it comes to actual work, as illustrated by the accompanying diagram, showing heat distribution, the total economy on the plant is as high, if not higher, when gas rather than steam is used, due to the fact that the sensible heat of the exhaust is largely used to raise the temperature of the blast under the producer, while at the same time the fixed ignition point of the very uniform gas so generated makes the timing of the engine far more accurate than with the usual form of composite

producer gas, which contains a varying amount of hydrogen as well as the usual amount of carbon monoxide.

2 I have also found in practice that no method of regulating the steam supply results in a uniform hydrogen content in the gas generated, due to the fact that the fire decomposes a varying portion of the steam supply used in proportion to its actual temperature, ranging from complete disassociation at 2000 deg. fahr., the normal temperature at full load, to less than 25 per cent disassociation at low loads, the balance of the steam being merely superheated and passing up through the fuel bed, finally condensing in the scrubber.

3 Secondly, it is almost impossible to use a simple fan exhauster, such as Mr. Lummis describes, and obtain a uniform pressure on the gas (regardless of the load on the engine), unless a water seal by-pass is used, as described in my paper; and further, a fan exhauster is not advisable in this connection as it is not positive in its action, and any increase in density of the fuel bed in the producer will affect the pressure on the line, as is not the case with a rotary positive exhauster and water seal by-pass. As stated in my paper, however, it is not necessary to put the gas under pressure with a single engine, this being a device gotten up to deliver gas to two engines operating in parallel from one suction producer.

4 As far as hydrogen content of the gas is involved, the same argument holds good in reply to Mr. Smith, but he achieves a uniform decomposition zone temperature in his producer by the internal design of his apparatus, enabling him to count on decomposing what steam he puts in, and in this way he is able to control very largely the percentage of hydrogen generated.

5 The large percentage of inert nitrogen referred to has not been found detrimental from an economic standpoint, as the expansion curve obtained in the engine with this diluent in the gas is perhaps more nearly perfect than where a gas containing hydrogen is used. In other words, it approaches more nearly the ideal air card.

6 Mr. Coleman's application of exhaust or burnt gases for the control of a producer supplying gas for an open-hearth furnace is a matter entirely distinct from the engine proposition covered in my paper, and my opinion would be that the decrease in the thermal value of the gas would be detrimental for furnace work where the main object is to obtain a high heat, there being no such compensating results as are obtained in the power proposition. This application has been successfully made, however, to my knowledge in connection with gas bench operation, and I understand that by careful design and effi-

cient methods, economy is shown over other methods of producer operation for furnace work.

7 I would say to Mr. Davison that my investigations have been confined to anthracite coal, but that I think the process should apply equally well to bituminous fuel, and when I have accurate data to offer on this subject, I will be glad to submit them to him.

8 Mr. Peck's remark that the greatest drawback to the producer gas proposition is its unreliability is undoubtedly true, and my investigations have been entirely directed to taking care of this defect. The solution of the problem, arrived at after considerable investigation, lies in the generation of an absolutely uniform quality of gas with a fixed ignition point, and with such a gas the gas engine is every bit as reliable and vastly more economical than any steam plant.

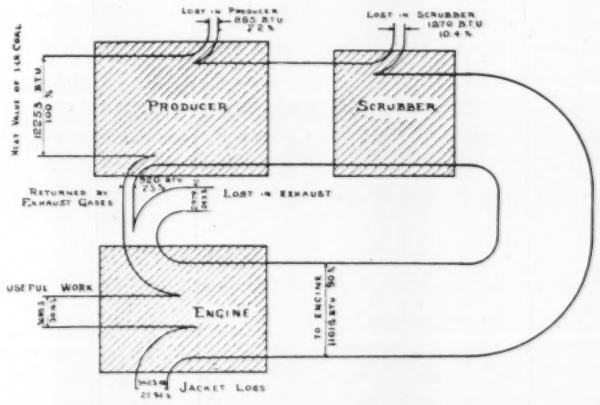


FIG. 1 HEAT DISTRIBUTION USING CARBON DIOXIDE AS AN ENDO-THERMIC AGENT INSTEAD OF STEAM

9 Unfortunately, too much attention has been paid to economy, and too little to reliability, but the results obtained with an equipment such as described in my paper were entirely satisfactory as to reliability, ease of operation, and attendance required for continuous operation 24 hours per day.

10 Mr. Stillman is undoubtedly right as to the type of man best adapted to run a gas engine plant; this has always been my experience.

11 It is pleasing to note that Mr. Jones has carefully inspected one of the plants referred to in my paper, and finds same operating

continuously and reliably with a minimum attendance, as I claimed, one man operating two engines, and one producer of 175 h.p.

12 Replying to Mr. Bibbins' discussion: From many gas analyses, taken on various producer plants, I have satisfied at least myself that the variation of the gas in quality and ignition point is the only variation noticed, and that when a gas of fixed ignition point and practically fixed heat value is generated, no matter how lean it is, entire reliability of operation is assured. Of course I understand that with a throttling governor the variation in compression pressure will also affect the ignition point; but this variation is a fixed one exactly proportioned to the load on the engine, and can be taken care of by automatic adjustment of the ignition mechanism for various loads through some such mechanism as that employed by Mr. Rathbun on his engines.

13 As to the action of ignition of a composite gas in a gas engine cylinder under compression, it would be difficult to prove conclusively that the more rapid gases burn out first and the more sluggish last, as the whole operation of combustion is so extremely rapid as to be impossible of observation, but I have every reason to believe that such progressive action does occur. It is in fact allowed for in using explosives such as gun powder and nitro-glycerine, where powder of various qualities is mixed together in the charge with a view to propagating the ignition uniformly through the mass, producing a rapid detonation; for example, wet and dry cotton are used in conjunction in torpedo charges to attain this end. The mechanical effect is an average rate of flame production corresponding in a way to the average ignition point of the elements used; and the quicker burning gas or explosives may be considered in this illustration as acting in the capacity of a detonator.

14 As Mr. Bibbins says, the adjustment of the ignition on a gas engine is a relatively simple matter if one knows what to expect from the mixture used, in the way of flame production, but to attain this production with any degree of perfection, the gas should have an absolutely uniform composition and ignition point.

HORSE POWER, FRICTION LOSSES AND EFFICIENCIES OF GAS AND OIL ENGINES

By Prof. Lionel S. Marks, Published in May Proceedings

The Author The brief abstract of the German discussion on the mechanical efficiency of gas engines, supplied by Mr. Suplee, shows

the confusion which exists at present in the statement of the power measurements and efficiencies of those motors. The vote at the Detroit meeting for the appointment of a committee to formulate a code of rules for testing gas engines gives the Society the opportunity to prevent that confusion in American practice. The suggestions which I have made, if accepted, would establish a practice different from the current practices of both Germany and England. It would make the statements of gas engine performances perfectly definite, practically and scientifically useful, and immediately available for comparison with other engines.

GAS POWER SECTION

COMMUNICATION FROM THE CHAIRMAN OF THE MEETINGS COMMITTEE

We believe that the work of this Section can be undertaken and carried out with more exact plans than have heretofore characterized Association or Society work.

2 We believe that the progress of the work of this Section will be largely determined by the wisdom displayed in planning its work. None of our members would attempt to do any construction work without first making complete plans for the execution of this work. An architect would not start to build his foundations until his completed plans were ready. While the method used in the execution of construction work may not be a true example of how the work of this Section should be undertaken, yet it is applicable to a considerable degree; and while we cannot at this date make complete plans and specifications for the execution of this work, we believe we can greatly facilitate the securing of results by suggesting a plan and submitting it to our members for their criticism and suggestion, both written and oral.

PURPOSE OF OUR ORGANIZATION

3 We are organized primarily to assist the development of gas for power purposes, and to assist in the progress and better the condition of the members of our fraternity. To accomplish our first object we must not only be able to influence all our own members, but must be able to influence all power users and their advisory engineers. To assist in the progress of our members we must not only furnish all possible information, but must maintain a sufficient interest in the subject to spur our members on to the greatest possible degree of achievement.

This communication is here presented both to the members of the Meetings Committee and to the membership of the Section for discussion and comment, as owing to the limitations of time it has been found impracticable to secure action upon it in advance.

UTILIZATION OF OUR TIME TO THE GREATEST POSSIBLE ADVANTAGE

4 At best, we will probably have not more than fifty hours in session for the consideration of these subjects during the first year. It is therefore important that we should use the time at our disposal to the greatest possible advantage. The problem is not simply to expend this time to the advantage, but, by wise discrimination, to expend it to the greatest advantage, of our members. We should frame our proceedings to bring out the most important matters as soon as possible.

DEGREE OF KNOWLEDGE POSSESSED BY OUR MEMBERS

5 In all probability, the knowledge of this subject possessed by our different members represents practically every state, from that of the novice to that of the scientist who possesses an exact knowledge of the present state of the art.

6 Manifestly, for lack of time we cannot start at the most elementary stage of this work, neither can we assume that a knowledge of the complete state of the art is possessed by sufficient members to ignore entirely the treatment of matters of which an understanding could be secured from other sources. It seems wise to assume for all a degree of knowledge readily acquired from other sources. Neglecting these then, our plan would be so to frame our program as to give our members a broad understanding of the whole subject, covering all fundamental information as fully as possible. In this way we can find wherein our members disagree, and by wise discussion eliminate many of the misunderstandings which may now exist.

ASSISTANCE NEEDED BY THE MEETINGS COMMITTEE

7 Your Committee admit their inability to determine many of the matters necessary to the formation of proper plans for the conduct of our meetings, and therefore wish to adopt the method of submitting them to the members directly, setting forth their present plans in sufficient detail to excite both suggestion and criticism, and by discussion of each section of this report to determine more nearly some of the more important matters under consideration, such as (1) misunderstandings now existing; (2) points of greatest importance and of widest application; and (3) points of importance overlooked by your Committee. The discussion of the Committee's report, both written and oral, will give your Committee much valuable informa-

tion, and we believe will do much to excite interest on the part of our members and give them valuable suggestions on how to furnish contributions of merit to the work of the Section.

THE REPUTATION OF GAS POWER

8 Should we fail to recognize the decided prejudice of many engineers and power users against gas power we would be unable to take proper steps for the removal of one of the greatest obstacles now retarding the more rapid adoption of gas for power purposes. It would be idle to maintain that no failures have been made in this field. We can however call attention to the fact that even repeated failures do not prove the actual limitations, but that a single success does prove the possibility of again securing at least an equal degree of success.

9 More discussion relatively has been given to the failure of some of the gas power installations than to the successes of others. This is not unnatural. The failure of the Quebec Bridge brought many more columns of publicity than all the successful bridges erected in the past ten years. Yet this failure no more proved that bridges cannot be used than the so-called gas power failures prove that gas power cannot be used. Many alleged failures have been due to a misunderstanding of the advantages and limitations of gas power.

10 Probably no better contribution could be made to the more rapid development of gas power than to give the power-using public a complete list of plants already in operation. We therefore desire to collect such a list, with a description of each plant and a record of its performance. We wish suggestions from all, of the names of those able to furnish information on this subject, as well as suggestions as to the best man to gather up this information from our members and give it to the Section in proper form at an early date.

11 We not only wish to secure reports of the actual experience of our members, but we also want reports which reach them indirectly. We want particularly to collect logs of tests and evidences of reliability as indicated by long and continuous runs, records of troubles and failures, and, as far as possible, an explanation for the causes for any trouble or failure which has been experienced.

STATE OF THE ART AS REFLECTED BY LITERATURE

12 The work we are engaged in is peculiar, in that much progress has been so recent and so much progress is being made every day that

the text books cannot be depended upon to reflect the true state of the art.

13 A complete bibliography would undoubtedly prove of great benefit to the work of the members along this line. It is probable that much of it is reflected only in various patent papers perhaps of recent date. A bibliography of text books and articles, with a short digest of each, and a list of patents described with sufficient minuteness to explain their nature, would be of special value. Some private records of this character are in existence although probably incomplete. Possibly some of our members have either a complete bibliography of the subject or a complete list of the patents.

14 Your Committee suggests, as a first attempt to cover the work, asking all members to place their records at the disposal of the Section. The question of editing and digesting this material is yet to be arranged in the office of the Secretary, but should include each month a bibliography for the month, with a short digest of articles appearing on the subject, and a list of patents granted, with a sufficient digest to indicate their character, and also each month a complete bibliography of the last preceding month not covered in the same manner. The idea is to get the current information and to work backward as rapidly as possible in order to secure complete information. We believe that we can thus cover the most important information first, and not being able to determine the amount of the task, we must not map out so much work as to be months, or perhaps years, in reaching the most recent information.

15 We thus hope to secure from our members, on this subject: (1) a knowledge of bibliographies now existing; (2) existing lists of patents; and (3) the names of the men able to give some of this information, or possibly available for consideration as a volunteer or employed editor.

THE STATE OF THE ART IN APPLICATION AND EXPERIMENTAL WORK

16 It is doubtful whether even the literature on this subect reflects the state of the art. There is always something, which never seems to be explained, which occasions temporary failure in all new undertakings, and can only be characterized as craftsmanship.

17 In gas power work different principles are involved in the type of engine used; for instance, the different cycles in the engines, different principles in carbureters and producers, and indeed in every portion of equipment and almost every detail in each portion of this

equipment. Any of our members versed in the practice pertaining to any portion of this equipment can, by volunteer service, make valuable contributions to the work of the Section and materially aid in the more complete understanding of the present state of the art, not only enabling us to determine the best of present practice, but how still further to improve it.

18 The results secured by some plants in operation, compared to those of others of the same character, indicate a refinement in craftsmanship which, if properly analyzed and described, would be of much assistance in securing better results elsewhere. The field offered for such a study of both manufacture and operation is sufficiently large to offer every one of our members an opportunity to make some valuable contribution to our work.

19 Your Committee makes no specific recommendations other than the request that each member consider carefully whether he cannot make some such valuable contribution. We can at least impress upon our members by discussion the desirability of rendering assistance to any other member endeavoring to furnish us such information. A proper degree of coöperation among our members can do much to procure and promote responses to inquiries made by an individual on behalf of the work of the Section, and we ask our members to give preference to those inquiries not made only for private reasons.

BUREAU OF ADVICE AND INFORMATION

20 We recommend the direction of the attention of members to the advisability of establishing means of assisting individual members by competent technical advice and the benefit of the experience of others. A knowledge of the experience of our members and what men have specialized on certain subjects will help us at least to direct inquirers to the men capable of giving them assistance.

21 The American Gas Institute has a bureau of information composed of competent men willing to give a certain portion of their time to the assistance of others in need of help.

MEANS OF COMPARING STEAM AND GAS POWER PLANTS

22 We also recommend a comparative analysis of the sources of loss in steam and gas power plants, showing points capable of direct comparison and the degree of each loss in each type; this analysis to cover the fundamental differences between the two, calling particular

attention to the following points, which we cite simply by way of example.

23 The particular points referred to, owing to a lack of understanding of the subject, have frequently been responsible for the failure to secure the high economy expected from the use of gas power.

a A gas engine has its highest efficiency at its maximum load, a steam engine at some intermediate load, both being rated practically at their point of highest efficiency. Since the steam engine has a nominal rating greatly below its maximum capacity, it therefore has a greater reserve capacity and a much less nominal capacity is required than for gas engines. It is also apparent that on a varying load the steam engine, by its ability to carry both over and under load, can be kept operating more nearly at its point of highest efficiency than the gas engine, which must always be operated at or below its most economical load, thus practically permitting variations only on the side of under load from an economical standpoint. This feature of the gas engine not only adds materially to the investment charges, as compared with the steam engine on the nominal rating of each, but also adds materially to the operating cost of the gas engine in relation to the greatest degree of economy attainable.

b The greater economy of operation will not offset the increased investment charges on large size gas power plants operating both on cheap fuel and a poor load factor, and as these charges are fixed in character, and greater for gas power than for steam power plants, the number of horse power hours generated per year in relation to each horse power of capacity will determine whether the lower operating costs will more than make up the increased investment charges so that the commercial efficiency will be greater for the gas power plant. On continuous operation each horse power in capacity would be able to turn out 8760 h. p. hours per year on a 100 per cent load factor and only 876 h.p. hours on a 10 per cent load factor. The cost of fuel will determine the load factor necessary to yield a higher commercial economy for the gas engine, and the lower the cost of this fuel the less will be the operating economy per horse power hour generated, and there-

fore the gas power plant is of less comparative value as the load factor and the price of fuel are diminished.

c As the gas engine has a much lower efficiency at points other than maximum load in relation to the consumption at maximum load, it is essential that the units selected permit of being run at a higher individual load factor, when running, than would be required for steam units; and this disadvantage evidently renders the gas engine relatively less suitable for classes of work where this high individual load factor on each operating unit cannot be secured.

24 The above points seem almost too simple to need attention, but they have been the source of considerable disappointment in the past; and we must bear in mind at all times that it is not sufficient simply that our members understand such matters; we must make them available to the power-using public that they may not fall into errors reflecting prejudicially on the adoption of gas power by others.

25 We are as much interested in setting forth clearly the disadvantages of gas power plants as their advantages. We not only wish to avoid further errors which unjustly reflect on their true merits, but we want matters of this sort understood that some of the disappointments which have been realized may be logically explained.

26 We recommend setting out fairly the points of merit and demerit of gas and steam power plants, bringing prominently into view the many advantages of the former, such as quickness and ease of starting, small standby operating charges, safety and availability, and ability to generate gas at one point and transmit it to a distant point for consumption—which is practically impossible with steam.

SPECIAL FIELDS OF USE FOR GAS POWER

27 We also recommend steps to bring prominently to the attention of our members and of the power using public the many ways in which gas engines can be used to secure the maximum of economy; such as the use of gas engines to carry that portion of a power load which is in constant operation, running the engines at their constant maximum output and using the old steam plant to take the load above this so-called "base load" and using the exhaust of the gas engines for the generation of steam or the heating of feed water for the steam plant. Many other special conditions where gas engines

are particularly desirable, could be cited, and it is hoped that all of them can be brought out.

UNPROMISING FIELDS AND UNNECESSARY DUPLICATION OF WORK

28 We believe that the treatment of these subjects will do much to eliminate a vast amount of unnecessary duplication of work now going on. Many experimenters in this field are trying to thresh out old straw. Time and money are being expended on features already covered in patent specifications. Experiments which have failed and perhaps their limitations demonstrated, are being repeated by others simply because the previous failures have never been reported. Difficulties overcome by some are still being experienced by others, due entirely to a lack of knowledge of the true state of the art.

29 We believe that much duplication work would be eliminated if the present state of the art could be quickly laid before our membership and provision made whereby all progress in the state of the art would be quickly reported. We also believe that by a broad and comprehensive treatment of the subject all members could more wisely choose the promising fields and the paths of least resistance.

30 No example is needed to convince all that much unnecessary duplication work is being done. That much work is going on of an unpromising character might be exampled by the consideration of the so-called gas turbine as distinct from what is known as the "mixed turbine," namely, a turbine using both gas and steam.

31 To construct a straight gas turbine successfully on ordinary lines requires the transformation of heat energy to work energy at a temperature and blade velocity beyond the power of any material now known to withstand. In spite of this fact, however, hundreds of hours and thousands of dollars have been wasted in the attempt.

32 On the other hand, many interesting fields are available which have not been fully called to our attention, such for instance, as gas making methods with the recovery of by-products. Two instances might be used as examples to excite interest in this subject.

a Mond, by the use of a great excess of steam, keeps the temperature in the producer at a point which protects the ammonia from disassociation and secures a yield of practically twenty pounds of ammonia gas per ton of coal used. Ammonia in the form of crude concentrate brings a price of from five to ten cents per pound of NH_3, or, by this method, a credit can be secured of one to two dollars per

ton of coal gasified other than for the lower thermal efficiency in this method of operation and the expense necessary to recover and concentrate the ammonia.

b Thousands of tons of coal are coked annually in bee-hive coke ovens, which waste all of the volatile products of the coal, including tar, ammonia and gas, excepting that which is usefully burned to carry on the coking process.

33 In spite of prejudice to the contrary, good metallurgical coke can be made in by-product coke ovens with the recovery of the same amount of gas, tar and ammonia as would be secured in the ordinary gas works practice. It is customary, however, to use a portion of this gas to heat the ovens, although producer gas could be used if desired; but even when using a portion of the gas yielded by the distillation of the coal, about fifty per cent of the better quality of the coal gas remains for use in other ways. This gas, by reason of its high hydrogen content, and also the presence of illuminating hydrocarbons, is not the most desirable for gas engine use.

34 By scrubbing with tar oils, the illuminating hydro-carbon vapors, which have too high a value for use simply as a fuel, may be recovered, and the remaining gas can be used in gas engines to generate power in the form of electricity, which can be transmitted long distances at a minimum cost of transportation. This gas, when denuded of its illuminating vapors has a value of perhaps 90 per cent per million B.t.u. of the value of the best gas which can be prepared for gas engine use.

35 While these two examples do not by any means cover this field, they are enumerated for the purpose of showing the importance of considering this branch of our work and also to excite further suggestions and exploitation on the part of our members.

MARINE PROPULSION

36 Probably no branch of the field has made more progress in the imagination of the engineering fraternity than that of marine propulsion. Naval officers of almost every country are interested in this subject and some interesting demonstrations have been made. We recommend an early statement of the development in this line, clearly setting forth its advantages and its special points of interest, not only to acquaint the public with what is being done, but to excite renewed interest in this promising field.

TRAIN PROPULSION

37 The success already demonstrated by the propulsion of vessels would indicate that gas power is also worthy of more serious consideration for train service. A start has already been made on this line of work in the driving of unit cars, generally using volatile mineral oil as fuel.

38 The problem in train propulsion by means of the internal combustion engine and solid fuel has not excited the same interest as marine propulsion. In fact, gas power adapts itself much more readily to marine propulsion, the power required bearing some exact relationship to the speed of the vessel in relation to the mediums (air and water) through which it passes, while the grades of railroads in conjunction with the weight of the load present complications requiring maximum capacities and introducing poor load factors which do not exist in marine propulsion—but in spite of the example just given and other disadvantages, train propulsion is worthy of serious consideration.

COMPARISON OF INTERNAL COMBUSTION ENGINES OF VARIOUS SIZES AND DIFFERENT TYPES

39 We recommend special attention to the different characteristics of internal combustion engines of various sizes and types, such as automobile engines and standard engines used on different characters of gases and different classes of work. From such a comparison we believe that progress in certain types, sizes and characters would contribute to progress in other branches.

40 It is interesting to note that the bids and specifications submitted by the manufacturers of large standard gas engines showed an average weight of about 700 lb. per h. p., while the average weight of all the automobile engines which took part in the Vanderbilt Cup Race was about 6¼ lb. per h.p. One of the engines intended for use in this race had developed 220 h.p. on a time run. Such a difference in weight without a wide difference in capacity would indicate that large sized standard engines could be built amply strong with less weight, less cost, and higher operating economy. Unnecessary weight, in this as in most instances, is a weakening factor and often a great obstacle to economy.

DIFFERENT METHODS OF MAKING GAS

41 We also think that different methods of producing gas can be advantageously treated, owing primarily to the fact that many gas power plants use gas for other purposes than the production of power, and this fact may necessitate a gas of special characteristics which would not be chosen if it was desired to use the gas only in internal combustion engines. In Europe, for instance, large quantities of blue water gas are used for welding purposes. This gas yields a higher flame temperature and has greater oxygen reducing tendencies than ordinary producer gas.

CLEANING GAS

42 Great difficulty has been experienced in cleaning certain kinds of gases. The cleaning of blast gases has been fairly well accomplished, although no doubt great improvements might still be made. The cleaning of producer gas made from certain bituminous coals has been exceedingly difficult, and in some instances has not been accomplished. The importance of this subject warrants separate and thorough treatment.

MEASURING AND TESTING APPARATUS AND INSTRUMENTS

43 Many errors have been due to faulty methods of measuring and testing. Some of our members are not even familiar with the standard apparatus now on the market for this purpose. A treatment of this subject, showing methods of measurement, including the Pitot Tube, the Venturi Meter, the old wet gas station meter, the different forms of proportional meters, the rotary meter, and the twin gasometers, would probably accomplish much good.

44 The question of calorimeters has already received consideration.

45 Methods of analyzing gases to secure greater speed and accuracy would also prove of great benefit. As an example, the Tuttle improvement of the Orsat apparatus makes three determinations possible almost in the time formerly required for one. Many other little wrinkles can be introduced, contributing both to speed and accuracy. For instance, by the use of an extra pipette—having two pipettes of cuprous chlorid—the partially exhausted reagent can be first used for the absorption of the CO, keeping the cuprous chlorid in the other pipette comparatively fresh and capable of greater absorption and quicker work.

CONCLUSION

46 Most of the gas associations have what is termed a "Wrinkle Department," to which all members forward any new plans or apparatus which prove of benefit in their work. The term "wrinkles" is assigned to these as generally covering improved apparatus of a simple but clever character which could hardly be dignified by the name of invention and yet of much practical value. This department, which has always attracted much attention in the gas associations and has furnished opportunity for the display of much ingenuity, has proved of great practical benefit.

47 The National Electric Light Association has adopted the policy of having a "Department of Progress," in charge of an editor whose duty it is to collect all possible reports of progress, summarize them, and present them to the members of the association from time to time.

48 We believe that both departments could be established in the Gas Power Section to advantage.

49 If the plan outlined above is carried out, your Meetings Committee must have the unqualified assistance and voluntary coöperation, and the generous contributions of the members. We must be universally assisted in finding where and how we can get this information into our proceedings. We suggest that in determining the best method of treatment for some of these subjects consideration be given to the adoption of topical discussions.

LOSS OF FUEL WEIGHT IN A FRESHLY CHARGED PRODUCER

By N. T. Harrington, Lansing, Michigan
Member of the Society

The object of the test was to determine the relation between the coal fired and that actually consumed during the first few hours of operation, after completely filling the producer with a new charge of fresh coal. The apparatus used was a suction gas producer, and gas engine arranged as shown in sectional drawing in Fig. 1.

2 The generator shell was mounted on platform scales, that the loss in weight due to combustion might be read directly. Its gas delivery pipe was connected to the scrubber by means of a water seal to allow free movement of the generator with the platform of the scales. All connections were flexible. A prony brake was used on the engine and the load was kept as constant as possible.

3 The coal was analyzed from the pile, and from samples taken as fired during the actual test. The ash as drawn from the producer, and the salvage as returned to the producer, were also sampled and analyzed. The gas was analyzed every hour. The final charge withdrawn from the producer at the end of the run was analyzed to show the increase in the amount of ash.

4 The test was started by weighing the producer empty and clean. The fire was then started and the producer filled with fresh coal, after which its weight was again taken. After blowing hot with compressed air to obtain good gas, the fire was poked enough to ensure a compact fuel bed, and the producer again filled. The engine was then started and the weight of the generator taken simultaneously.

5 Throughout the run all fresh coal added to the producer and the ash and salvage taken from the producers, were weighed on separate scales. The fire was cleaned and the producer refilled every 3½ hours. The loss in weight of the fuel in the producer and the brake load on the engine were noted every 15 minutes. All weighings of producer and charge were so made and calculated that the

consumption of combustible could be recorded every 15 minutes, correction being made for the ash and salvage drawn from the producer, and the salvage returned to the producer.

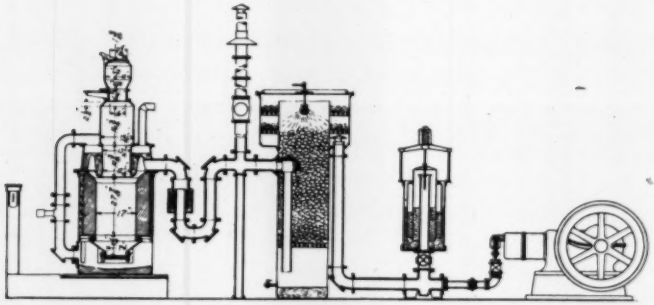


FIG. 1 SECTIONAL DRAWING OF SUCTION GAS PRODUCER AND GAS ENGINE SHOWN BELOW

6 The water in the vaporizer, and the water in the seal between the generator and scrubber, were kept at constant levels.

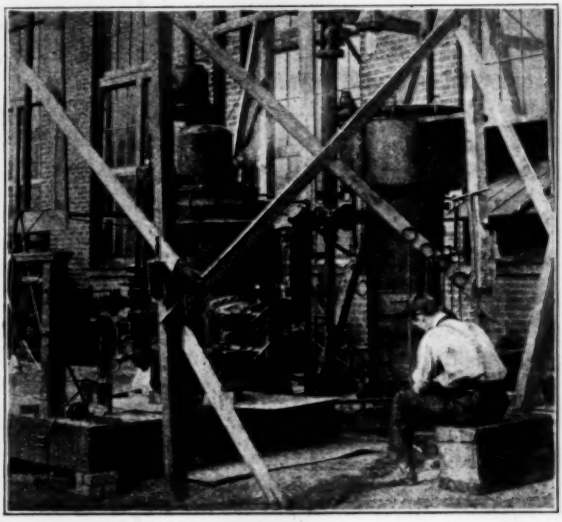


FIG. 2 SUCTION GAS PRODUCER AND GAS ENGINE

7 At the end of the run the fire was cleaned and the producer filled, after which engine and producer were shut down and the final

weight of producer and charge taken immediately. The entire charge with the exception of the clinkers sticking to the walls was then withdrawn from the producer and quenched, and the producer again weighed to determine the amount of clinker sticking to the walls. The clinkers were then removed, pulverized, and mixed with the rest of

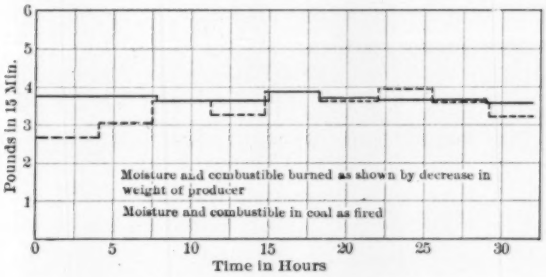


FIG. 3 RECORD OF COMBUSTIBLE AND MOISTURE

the charge. The whole was then sampled and analyzed. By this means it was possible to strike a balance between what was put into the producer and what was taken out.

8 Fig. 3 shows the relation between the combustible burned during the nine periods of the test, corresponding to the nine times the fire was cleaned and producer filled; and the combustible fired

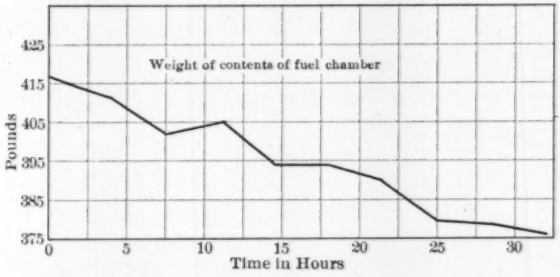


FIG. 4 RECORD OF LOSS OF WEIGHT OF CHARGE

to the producers during the same periods. Figures for combustible were taken to include moisture. The combustible burned was determined by the loss in weight shown by the scales under the generator. The combustible fired was determined by deducting the ash contents from the weight of coal as fired, and weighed separately on small scales.

9 While these two curves do not show the actual amounts of coal fired and consumed, it may be assumed that the ordinates are directly proportional.

10 Fig. 4 shows the actual loss in weight of the charge in the producer by periods.

11 Care was taken that the fire was clean and without cavities at the times of filling, and that no arches or stoppages occurred in the fire at any time during the test. At the end of the run the fire was poked vigorously from the top to determine how much additional coal could be jammed into the producer. Only 10 lb. more could be added, and it is possible that this slight settling of the coal was due to a portion of the charge being forced up into the gas ring.

12 The difference of 40 lb. between the weight of the charge at the beginning and end of the run is to be attributed to the burning out of the carbon and volatiles from the lumps of coal, leaving the laminated ash structure to occupy about the same space as the original lumps of coal but increasing the ash contents of the mass.

13 For a short run with a fresh fire, it would seem that the coal fired to the producer does not represent the actual coal consumed and that in a 30-hr. run it may be as much as 8 per cent too low.

GENERAL DATA

PRODUCER

Duration of test	32 hr.
Diameter of fuel body	16 in.
Area of section of fuel body	1.395 sq. ft.
Weight of generator empty	3491 lb.

ENGINE

Single cylinder, single acting, four cycle

Diameter cylinder	$7\frac{7}{8}$ in.
Stroke	$10\frac{1}{2}$ in.
R.p.m.	380
Average b.h.p. developed	12.6

COAL ANALYSIS

	Percentage coal as fired	Percentage final charge drawn from producer	Percentage ash
Moisture	2.58	.....	.10
Volatile Carbon	5.47	2.72	1.68
Fixed Carbon	74.09	75.49	28.42
Ash	17.86	21.79	69.80

RESULTS

Weight of charge in producer at the time of starting run, pounds	416	Loss of combustible and moisture by weight during run, pounds	471.15
Weight of coal added during run, pounds	506	Weight of ash withdrawn during run, pounds	76.75
Total	922	Weight of final charge withdrawn from generator, pounds	376.00
Error	1.9		
Total, pounds	923.9	Total, pounds	923.90

Weight of charge in generator at beginning of run, pounds 416
Weight of final charge withdrawn from producer, pounds 376

Loss in weight of charge due to fire loading up with ash, pounds 40

BITUMINOUS PRODUCER PLANTS

By Elbert A. Harvey
Non-Member

The process used in the plants here described involves the generation of producer gas in updraft producers without any attempt to burn the heavy hydrocarbons or tars into gas in the gas generators, the hydrocarbons being removed from the gas by a mechanical cleaning process carried on in a rotary gas washer, leaving the gas clean and in possession of its light hydrocarbon constituents, and therefore a far richer and more useful gas than can be produced by any downdraft or back draft process in which the tars are burned into gas in the producers.

2 The apparatus used for this process, which is known as the Industrial Gas equipment, includes also apparatus for the recovery and utilization of the tar removed from the gas, deriving from this the power required for the operation of the gas washer and the necessary steam or vapor for generating the gas. The energy of the coal is thus separated into three parts; one in the form of chemical energy represented by the final product, a clean cool producer gas, and amounting to about 75 per cent; the second in the form of tar and amounting to about 12½ per cent; and the third in the form of sensible heat lost in radiation from the gas generators and carried away by the cooling water, and amounting to about 12½ per cent.

3 The producer is of the updraft water-sealed type developed directly from the original Siemens producer, and today used in

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identically the same design (but not in connection with cleaning apparatus) with open hearth and glass melting furnaces, and for a great variety of processes requiring raw uncleaned producer gas.

4 The Rotary Gas Washer is the direct outgrowth of a type of machinery of which perhaps the most primitive sample is the farmer's churn, the immediate antecedent being the rotary dust cleaner for blast furnace gas which had its development in Germany; and the tar burning outfit is simply a variation of fuel oil equipment. On the other hand, the application of these various equipments to the production of clean producer gas from bituminous coal involves new features and produces good results.

5 In seeking, like many others, a substitute in the event of shortage in their natural gas supply, the management of an Indiana Glass Manufacturing establishment conceived the idea of using some of the raw bituminous producer gas which they were burning in their gas melting furnaces. Their producer equipment was similar to that used in the large glass factories throughout the country, in which the gas is conveyed in large brick conduits six to eight feet square or in diameter. The gas is under only the slightest pressure and rolls along like heavy smoke depositing the tar and soot. The flues are burned out once a week to free them from the tar and soot.

6 After some experimenting at home, these gentlemen investigated foreign gas producer equipment, and Mr. Gustav Saaler, a German engineer of reputation, designed a gas cleaning equipment for their use. Today the company are operating their gas power plant on producer gas from Indiana and Pittsburg bituminous coal, and developing an average load of 1320 b.h.p. on two Koerting double acting two cycle gas engines, direct connected to two 400 kw. Westinghouse a.c. electric generators connected in parallel. These engines are rated at 600 h.p. each for natural gas, but on producer gas are successfully carrying this overload 24 hours per day, 6 days per week. The present gas power plant of this concern embraces the following equipment:

Four Taylor gas generators, 10 ft. diam., remodelled in 1907 by the Industrial Gas Company of New York so as to include the Herrick tuyeres.

Two Saaler rotary gas washers, with accompanying coolers and dry scrubbers, installed in 1906–1907.

Three 600 h.p. Koerting double-acting two cycle gas engines, operating at 100 r.p.m., direct connected to 400 kw. Westinghouse a.c. electric generators connected in parallel,

two engines being in constant service with one spare idle unit.

One 150 h.p. Westinghouse vertical three cylinder gas engine, direct connected to d.c. electric generator for lighting purposes.

7 Some gas is used in their rouge ovens (used for gas polishing). Tar outfit will be installed later.

8 This power plant has been operating for over a year and has made a complete continuous run during the year of 1907, 24-hour service most of the time, furnishing power in a satisfactory way to meet the local factory conditions. Complete statistics of this plant are not available. The engines are showing excellent economy, averaging 10 573 B.t.u. per b.h.p. hour. However they have not been satisfactory in all particulars, and this type of engine has since been withdrawn from the market by the De La Vergne Machine Company, whose present engines, by the way, are showing considerably better fuel economy and better construction in other details. This company operates two power plants; one, the gas plant just referred to with its average load of 1320 h.p. and an average coal consumption of 15 tons Pittsburg No. 8 three-quarter lump per 24 hours; the other is a steam plant, average load 1250 h.p., average consumption of same coal 75 tons per 24 hours. The two plants present one of the most striking illustrations of the contrast between power plants of this size, both operating under the same factory conditions. This equipment cannot be considered the equal of what could be put in at the present time, but it has been one of the pioneer plants which has done yeoman service in establishing the reputation of the successful continuous operation both of the gas engine and gas producer plant.

9 While detailed statistics are not furnished for this plant, I am permitted to state that the record for the year 1907 showed a total coal consumption in the producers of 1.34 lb. of coal per kw. hr. at the switchboard, equal to 0.9 lb. coal per h.p. hr. on the driving wheels of the engines. These figures, however, do not take into consideration the fuel required to operate rotary washer and furnish steam to the producers, which increases these figures to 1.75 lb. of coal per kw. hr. at the switchboard and 1.06 lb. of coal per h.p. hr. at the engine drive wheels.

10 The figures for the auxiliary equipment are estimated, and these figures cannot therefore be presented as absolutely authentic, but they are the nearest estimate which I am able to make, and I

believe represent the actual condition. The following figures indicate the probable heat balances representing operating conditions at this plant:

PROBABLE HEAT BALANCES IN CONNECTION WITH 1800 H.P. GAS AND ELECTRIC POWER PLANT OPERATING 24 HOURS PER DAY, 6 DAYS PER WEEK AVERAGE LOAD 1320 B.H.P. AT SWITCHBOARD

COAL		
1.06 lb. Pittsburg No. 8, assumed average 14 249 B.t.u. per lb., giving a total B.t.u. per h.p. hour		15 104
PRODUCER LOSSES		
(1) Radiation: 10 per cent of heat value	1510	
(2) Tar: 8 per cent of heat value	1208	
(3) Power and Steam for gas making 12 per cent of heat value	1812	
GAS ENGINE LOSSES		
Engines using 10 573 B.t.u. per b.h.p. hour: 51.2 per cent of heat value	7746	
ELECTRICAL GENERATOR LOSSES		
Assuming 90 per cent efficiency, making 1.9 per cent of heat value	283	
Total losses		12 559
One theoretical h.p., showing a net power plant efficiency of 16.9 per cent		2 545

11 The actual cost of power for this plant from the company's books is not available, but it is possible to offer fairly accurate figures showing the probable cost of their power based on conditions as they exist. The two principal factors are known, viz., the coal consumption and the number of men employed. The steam and power required for auxiliary equipment can be closely estimated, and other charges are based on standard practice. The figures are made up on the basis of the cost of 1320 h.p. average load during 24 hours for one day.

COST OF GAS POWER IN 1907 AT THE FIRST INDUSTRIAL GAS GENERATING AND CLEANING PRODUCER GAS POWER PLANT: 1800 H.P. IN THREE UNITS, TWO OPERATING AND ONE IDLE. SERVICE TWENTY-FOUR HOURS PER DAY, SIX DAYS PER WEEK. COST FOR ONE DAY—TWENTY-FOUR HOURS

COAL	
In producer, 15 tons per day at $2.50	$37.50

STEAM

35 h.p. for gas washer at 34½ lb. per h.p. ..... 1210 lb. per hour		
For gas making ..... 415 lb. per hour		
This makes approximately 45 h.p. per hour costing say 50 cts. per day		12.00

WATER

3 Gallons per h.p. hour in cooler		
2 Gallons per h.p. hour in rotary washer		
9 Gallons per h.p. hour in gas engine		
This figures 450 000 gallons per day, costing at a pumping charge of one cent per 1000		0.45

LABOR

One head gas man for each day and night shift, superintending the producer men and in actual charge of the gas washers, cooler and dry scrubber, at $2.50	$5.00	
Two producer men for each shift, charging coal, poking producers, and cleaning out ashes, at $1.50	6.00	
One gas engine engineer responsible for engine room both shifts, and in actual charge one shift	3.00	
One assistant gas engine engineer in charge of engine room, night shift	1.75	
Two engine room oilers and helpers each shift, $1.50 each	6.00	
One helper: the coal is brought to the charging platform, the ashes carted away, tar removed, and there is other intermittent work attended to at odd times by various laborers, making the full equivalent of one man's time each shift, at $1.50	3.00	$24.75
Oil and Waste		3.00
Light and Miscellaneous expenses		1.00
TOTAL OPERATING CHARGE PER DAY		$78.70

OVERHEAD COST

Assuming a total investment of $100 per h.p. for an 1800 h.p. Gas Power Plant, amounting to $180 000, and figuring 15 per cent overhead charge to cover interest, depreciation, maintenance, insurance, taxes and supervision there is a yearly fixed charge of $27,000, or per day	$90.00
TOTAL COST OF 1320 E.H.P. AT SWITCHBOARD FOR ONE DAY, TWENTY-FOUR HOURS, ...	$168.70
Cost per kw. at switchboard, 8/10 cts.	

12 Some months ago I had occasion to make up carefully figures on a complete gas-electric power plant of 20 000 h.p. for twenty-four hour service, full and steady load, 308 days per year. The following

figures give the summary of carefully worked up detailed estimates of the cost of equipment and cost of power:

ESTIMATED COST OF COMPLETE 20 000 H.P. ELECTRIC POWER PLANT, INCLUDING EVERYTHING UP TO CONNECTIONS FOR THE DEPARTING SERVICE WIRES

Gas Engines and Electric Generators	$780,000.00
Gas Producers and Auxiliaries	200,000.00
Building	60,000.00
Miscellaneous, including Piping, Pumps, Exciters, freight, etc	60,000.00
	$1,100,000.00

COST PER H.P. OF EQUIPMENT, $55.00

ESTIMATED ANNUAL COST OF ELECTRIC POWER AT THE SWITCHBOARD, 20,000 H.P., FULL CONTINUOUS LOAD, 24 HOUR SERVICE, 308 DAYS PER YEAR

COAL

For entire equipment, bituminous run of mine, at 1¾ lb. per kw. hour at switchboard, amounting to 84 000 tons at $2 per ton	$168,000.00

LABOR

Including chief and assistant engineers for both shifts, and all necessary assistants, amounting to forty men	$28,800.00
Oil, waste and Miscellaneous Supplies	24,000.00
Overhead charges at 15 per cent on $1,100,000	165,000.00
Total cost of 95 000 000 kw	$385,800.00

Cost per kw. hour at switchboord, 4/10 cts.

13 The above figures were made up on guaranteed efficiency of 70 per cent for the Producer Gas Engines operating on 10 000 B.t.u. per b.h.p. hr., 90 per cent for the Electric Generators, and the cost figures are to the extent of 90 per cent based on actual quotations.

14 The next plant installed was to replace natural gas in small furnaces for tempering, case hardening and light forging. This plant was started in the spring of 1908, and proved a complete success. The gas gave most satisfactory results in all their furnaces, producing heats as high as required with the gas delivered at a pressure of about 3″ of water and air from a fan at slightly higher pressure. No regeneration whatever has been used. This plant has demonstrated that producer gas can be used with entirely satisfactory results for work of this class not requiring high heats. Some test runs have been made here for which heats sufficiently high for forging and welding have been claimed, but these claims have not been demonstrated by actual operation but

are based upon judgments formed by the incandescence of the furnace. It is my opinion it will prove practicable to use producer gas of this kind in small furnaces for heavy forging and welding operations, but undoubtedly some regeneration will be required for this kind of work. As yet the point has not been thoroughly tried out, and claims are being made quite generally that producer gas can be used without regeneration or preheating for small furnace welding and melting operations. On the other hand it has been and is still to a large extent the opinion of gas engineers that even if such results can be secured by the use of high pressure air, it would not be economical to do so, and that the cost of equipment for regeneration would be too great to warrant the use of producer gas for this line of work, the conclusion being that producer gas cannot compete with high temperature gas such as water gas or natural gas in this field. The firm with which I am connected hopes within the near future to demonstrate this point in connection with gas of this kind under actual shop operations.

15 This second installation just referred to is operating on run of mine Indiana coal, locally known as Vandalia No. 4. This has proved an excellent gas coal, the average analysis of the coal and gas being as follows:

COAL			GAS	
Moisture	6.65	CO	21.47 at 324 B.t.u. = 70.	B.t.u.
Volatile H.c.	36.75	H_2	15.29 " 278 B.t.u. = 42.5	B.t.u.
Fixed Carbon	50.35	CH_4	3.58 " 919 B.t.u. = 33.	B.t.u.
Ash	5.40	O_2	0.00	
Sulphur	Not given	CO_2	8.57	
B.t.u.	" "	N_2	5.109	
			100.	145.5

16 In giving the heat value of the gas a minimum standard is adopted, viz: the gas is taken at 62 deg. fahr., 14.7 lb. pressure, and that portion of its heat value represented by the latent heat of the steam formed from the hydrogen and the hydrogen in the hydrocarbon is discarded. Without these subtracted, the heat value of the gas at the same standard of temperature and pressure would be 156 B.t.u. per cubic foot. This is excellent gas, and about the average for gas made in this process. It is considerably better than the gas made from poor slack coals which average from 120 to 130 B.t.u. per cu.

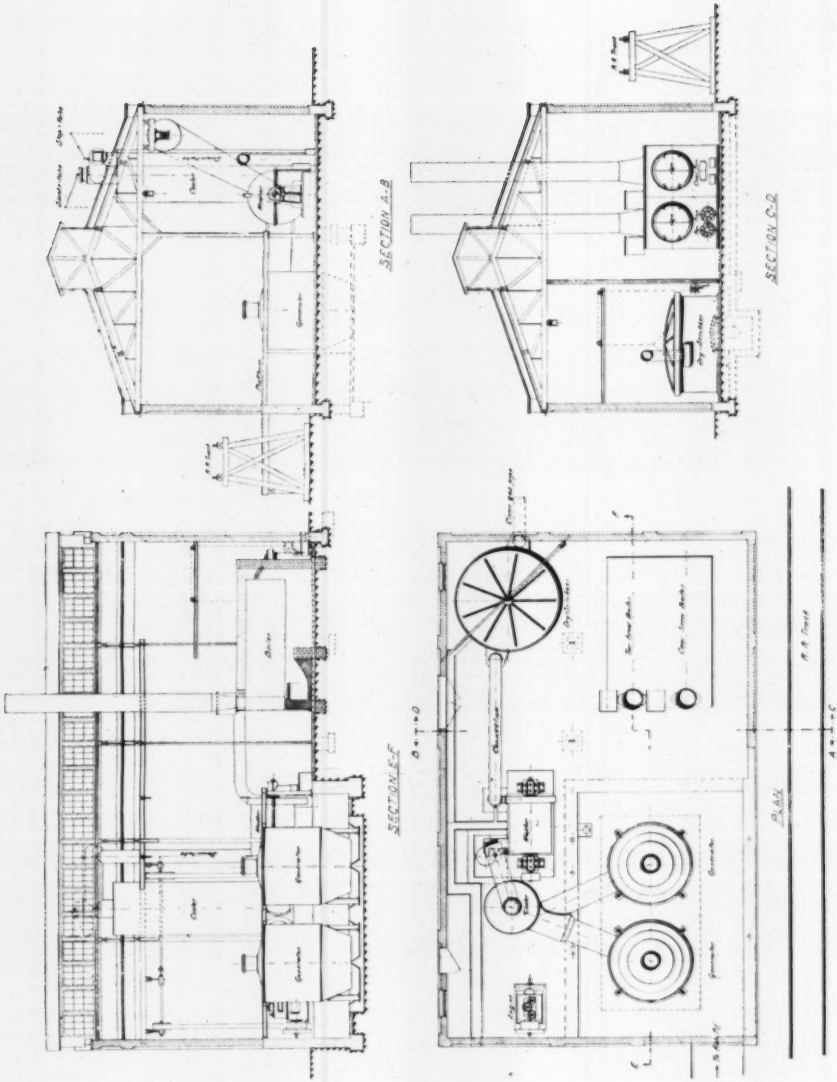

FIG. 1 2000 H.P. BITUMINOUS GAS GENERATING AND CLEANING PLANT

ft. and not as good as the gas made from lignite, which will average from 160 to 170 B.t.u. per cu. ft.

17 This plant has not yet been in operation long enough to establish accurate operating costs but a fairly close estimate of the cost of the gas is shown in the following table made up from the most accurate information I can get:

COST OF PRODUCER GAS MADE IN 2000 H.P. PLANT OPERATING IN CENTRAL INDIANA, INSTALLED IN 1908

DAILY COST OF THE EQUIVALENT OF 200 000 FEET NATURAL GAS IN 10 HR.

COAL		
Used in producer, Indiana run of mine, Vandalia No. 4: 10 hours operation, 17 000 lb., stand-by losses, 1000 lb.; making a total of nine tons at $2		$18.00
AUXILIARY FUEL		
Steam for engine operating gas washer, average load 60 h.p., using per hour	3200 lb.	
Producer using per hr	500 lb.	
Tar handling and burning outfit per hour	500 lb.	
Total steam per hour	4200 lb.	
This amounts to approximately 125 boiler h.p.		
They require when operating on coal only, 6000 lb. per day. The tar amounts to approximately 10 per cent in weight of the coal burned, or 1250 lb. per day, and each pound furnishes as much steam as two pounds of coal making the equivalent of 2500 lb.; net coal fired in boilers per day	3500 lb.	
Making 1¾ tons, at $2		$3.50
WATER		
100,000 gallons per day at a pumping cost of one cent per 1000 gal.		1.00
LABOR		
One gas man in charge, and also caring for washer, steam engine, cooler, tar outfit, and boiler	$3.00	
Two and a half helpers: two men on charging platform and the equivalent of half of one man's time for odd work, at $2	5.00	$8.00
OIL AND WASTE		0.50
OVERHEAD CHARGES		
15 per cent per annum on an assumed total cost for gas plant, gas house, charging platform, elevated track, furnace and pipe alterations, of $35 000, making an annual charge of $5250, or per day		17.50
Cost of the equivalent of 200 000 ft. natural gas per day		$48.50

Cost of equivalent of 1000 ft. natural gas, $0.2425.

18 This 2000 h.p. plant, which was the second to be installed, is a complete typical equipment, and will serve very well to show the design of this class of apparatus. The accompanying photograph shows the gas house. This outside view of the gas house calls attention to a point of first consideration, viz: coal delivery to the gas premises. An elevated track is shown coming in on a level some six or eight feet above the charging platform, or working platform on the producer tops. The coal is dumped from bottom dump cars into the bins shown under the track and runs right in on to the

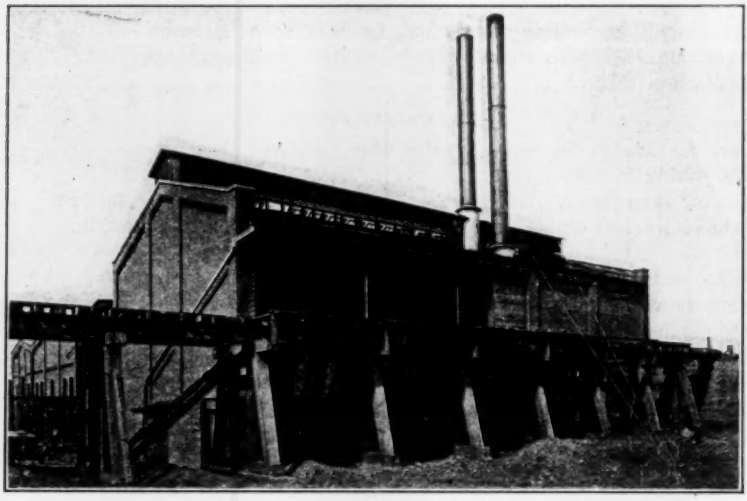

FIG. 2 GAS HOUSE, 2000 H. P. BITUMINOUS GAS PLANT

charging platform. Another view illustrates this feature, showing two producer tops, each with one coal hopper, and a number of poke holes.

19 The charging platform is of steel frame work with concrete filling. The producer tops are made up of heavy castings, studded with fire bricks, the bricks projecting down below the lower surface of the metal. This arrangement affords a strong cool top which does not require water cooling, and which is sufficiently tight for use with bituminous coal, as the tar in the gas soon fills up all leaks. It should be noted that the wall of the gas house is not built in over the charging platform, leaving this part exposed to the air. This is desirable because some gas escapes while the fire is being poked,

making it almost impossible for the men to work where the producers are located in closed rooms. This gas house was designed so as to provide the open or out-door feature for the charging platform and the protected closed power house for the rest of the apparatus.

20 In making gas from bituminous coal it is desirable to charge the coal as uniformly as possible for the reason that a large percentage of the fuel value of bituminous coal consists of volatile hydrocarbon represented in the final gas as CH_4 and C_2H_4. This is volatilized by the heat in the producer without any chemical reaction, and is given off soon after the coal is introduced into the producer. This

FIG. 3 STEEL FRAME CONCRETE CHARGING PLATFORM

process is quite distinct from the production of CO by the primary producer process of semi-combustion, and in order to keep the gas with approximately the same ratio of these gases it is necessary to introduce the coal into the producer at an approximately uniform rate. In practice it is found satisfactory both for gas engine and small furnace requirements to charge about once every fifteen minutes. For power plant work, where producers of large size are used, 10 or 12 ft. outside diameter, it is necessary to have one man per producer. These men put in the greater part of their time poking the fire to keep it in condition. The poorer the coal, the more poking is required. As these same men charge the coal, no more labor is required on account of this frequent charging than would be required otherwise,

and on this account there has been no attempt to adopt automatic coal feeders in connection with this class of apparatus.

21 The poke holes on these producers are all located on top and quite near the sides of the producer. This makes it possible to scrape the sides with the pokers and avoids the necessity of putting poke holes in the sides of the shells, and makes it possible to have a solid brick lining without any apertures, and therefore a lining that will last.

22 The entire gas making is controlled from this charging platform. The steam valves governing the supply of steam for the producers are located here, and the quantity of gas is regulated by these valves. All that is necessary to stop at any time is to cut off the steam, open one of the poke holes, light the gas and let it burn until the flame dies out; and in the morning all that is necessary is to turn on the steam which blows up the fire, let the gas escape through all the poke holes for awhile until the quality is seen to be good, and then close the poke holes and gas of good quality with the proper pressure will be found to be in the mains.

23 These producers are built as plain brick lined cylinders, as shown in the accompanying cut, without grates, and with the fuel body resting immediately upon the ash which extends down to the bottom of the concrete water basin below the producer, the producer being entirely open at the bottom, the water forming the bottom. Air and steam for gas making are introduced through tuyeres, or blast boxes, projecting through the sides, the air and steam getting into the fuel through slots in the sides and ends of these tuyeres and being thus distributed uniformly over the entire surface and passing in this way up through the fuel body. The ash is maintained at a level about one foot above the tops of the tuyeres and it is desirable to maintain at least three feet of burning coal above this, but to keep the coal sufficiently deep so the top of the fuel bed does not show very hot, being partially black. The operator soon learns to judge of the condition of his fuel bed and the quality of gas being made by the temperature as seen through the poke holes, and by the appearance of the gas escaping through the poke holes. If gas of a good quality is being made, the fuel bed appears at the right temperature, and the gas appears of a decided brown tinge; but if the temperature runs up or down, it means that an excess quantity of air is being introduced and part of the gas is being burned in the producer, raising the temperature, or an excess quantity of steam is being introduced which is lowering the temperature of the fuel bed, and if this continues the

fire will soon arrive at a stage where no gas whatever can be made. Any expert gas maker can judge his fire simply by the eye, but these producers are equipped with thermostats indicating the actual temperature in the producer necks, and with U-tube water pressure gauges indicating the pressure, so that the operator always has before him the exact temperature and pressure conditions, and can regulate his steam valves, coal feeding and poking accordingly.

24 The distribution of air and steam in these producers is so excellent that very little trouble is experienced with clinkers even with low grade coals, and so far as results are concerned these producers have been just as successful with slack Indiana and Ohio coals as with the best run of mine gas coals. However, the poorer the coal, the more labor is required and the smaller the quantity of gas produced, and the selection of coal must be determined by each individual manufacturer after balancing the costs of coal and labor. and frequently it is found that high grade gas coals are the cheapest even when secured at considerable advance in price.

25 In order to appreciate the significance of the various features of this gas generator, it is well to recollect the development of the producer and just what producer gas is. Producer gas was first introduced by Siemens in England in the 60's. The original producer was simply an attachment to a furnace in a way whereby combustion was carried on at two points, there being a primary and secondary combustion. The idea was to produce semi-combustion in the first chamber by limiting the supply of air so that the carbon in the coal would burn into CO instead of CO_2, and then introduce the remaining necessary air into the second combustion chamber to complete the combustion by burning CO into CO_2. Carbon gives up approximately 30 per cent of its heat in the first process, and the remaining 70 per cent in the second. By building the producer in connection with the furnace there is very little radiation loss, and this arrangement gave pretty close to a theoretical 70 per cent efficiency and accomplished a degree of control and continuous operation not obtainable in a hand-charged furnace, the principal reason for this being that the intermittent charging of the coal into the producer did not seriously affect combustion conditions in the furnace.

26 The next step in producer construction was to introduce a quantity of vapor with the air. This vapor was decomposed, giving off hydrogen in the producer, and in this way absorbed part of the waste 30 per cent given off by semi-combustion. This recovery of waste heat by the introduction of vapor is so effective that in a num-

ber of producers operating in this country today, delivering hot gas immediately to the combustion chamber, there is an actual working efficiency of over 90 per cent. Figures of this kind are published by the Morgan Construction Company, Worcester, Mass., on apparently valid authority.

27 Several Siemens producers and furnaces were imported and operated in this country in the 60's, but the first producer to gain any prominence in this country was a producer manufactured under English patents and placed on the market in this country by the Duff Patents Company, which company is today one of the most prominent producer builders in the field. This producer embodied all the points of the original Siemens producer, but added the feature of an open bottom standing in a basin of water, the water forming the bottom; the idea being that the ash could be removed under the surface of the water during operation, thus making it possible to operate continuously without shutdowns. In order to remove the ash in this way during operation, it proved necessary to use grates extending over only a part of the producer area, so that clinkers could be poked down and taken out below the water. This type of producer has for the past twenty years furnished gas to the great proportion of glass and steel mills. The principal difficulty which has been experienced with the Duff producer was the fact that the distribution of the air and steam was not uniform over the entire area of the fuel bed, and clinkers were apt to form and cling to the grates.

28 The next step in advance in producer construction was the introduction of the Talbot type of producer now represented by the producers of the Morgan Construction Company, R. D. Wood & Co., and others, which replaces the grate with a central mushroom blower, thus bringing a nozzle for the air and steam right up into the center of the fuel body. This accomplishes very much better distribution than in previous arrangements and has only within the last year or two been improved upon.

29 The Herrick tuyeres distribute the air and steam more completely, and the result is less trouble from clinker, increased capacity, and a great reduction in the labor required for gas making. Furthermore, with this uniform distribution it is possible to gasify many low grade coals without an unreasonable amount of labor. These producers are equipped with Eynon-Evans steam induction blowers, four tuyeres each with a blower being used on the 12-ft. diameter producers, and three each with a blower on each of the smaller sizes.

Each blower is supplied with a ½ in. steam pipe, the steam being delivered under a pressure of from 10 to 20 lb. A pressure regulating valve is placed at a convenient point for adjustment for the men on the charging platform and is adjusted from time to time so as to maintain a uniform gas pressure and at the same time deliver the supply of gas called for. These blowers draw in the air required for gas making by induction, the quantity of air being regulated by an air shutter, and when once adjusted need further adjustment only very occasionally.

30 The accompanying sectional cut shows a view through the entire producer gas plant. Gas is delivered from the gas generators

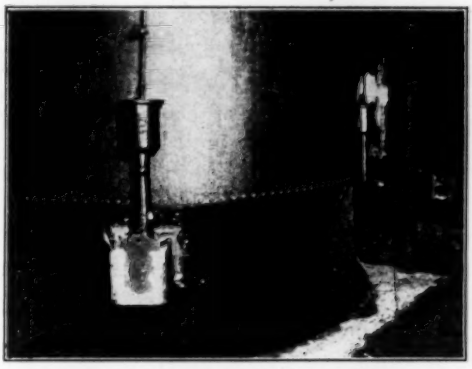

FIG. 4 HERRICK GAS PRODUCER

through brick lined flues into cooling towers. These are simply large steel tanks built with small diameter and proportionately large height; this apparatus for 2000 h.p. being 8 ft. in diameter and 30 ft. high. They are equipped with water sprinklers, providing for a shower of water passing down and striking against the flow of the gas passing upward. This cools the gas and removes a very small portion of the tar. The cooler has a sloping bottom terminating in a trough. This is arranged in a manner so that the water passes out at one level, the light tars float on the top of the water and pass off at a slightly higher level, and the heavy tars collect at the bottom where they are drawn off periodically. We are building these coolers now without bottoms, standing them in water basins, the water forming the bottom so as to prevent any obstruction or collecting point for the tar, and so as to facilitate the removal of the tar. The

cooler is also provided with several manhole doors at convenient points so that they can be scraped from time to time. It is not yet known just how often this piece of apparatus needs cleaning, but it can be stated, however, for the entire apparatus that there is no part which requires a shut down for cleaning except at long periods, so that a single unit apparatus can be built for continuous twenty-four hour service every day in the week, provided there are short shut-downs once or twice during the year.

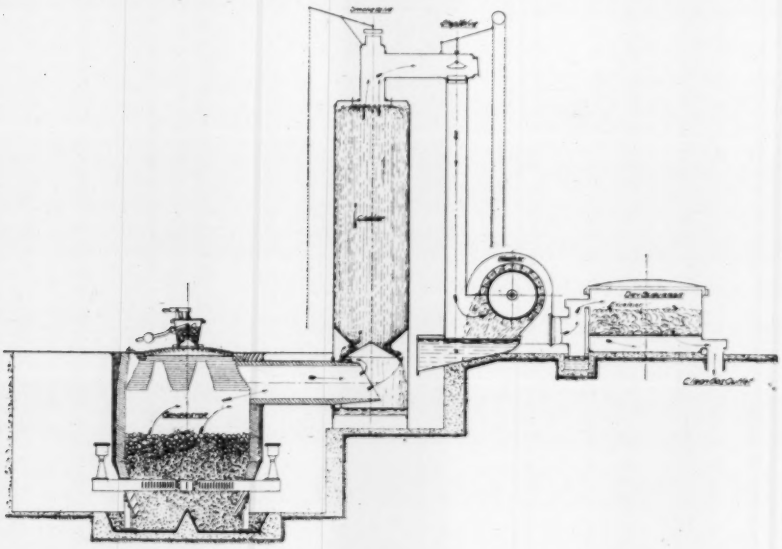

FIG. 5 INDUSTRIAL GAS GENERATING PLANT

31 The pipe leaving the cooler is shown with projection, with a cap valve operating with a lever handle. This serves the purpose of a stack and is used to allow the gas to escape when first blowing up the fires, as it is difficult to blow the fire up quickly if only the poke holes are used for this purpose. Frequently this stack is located between the producer and the cooler or elsewhere than shown here.

32 Gas is conveyed from the cooler to the rotary gas washer. Partial views of this machine are shown in accompanying photographs, and in the sectional view of the entire plant. Primarily this machine consists of a cylindrical drum encased in a shell of the same shape, both drum and shell having steel plate projections which clear

by about $\frac{1}{2}$ in. as the drum revolves within the shell. The gas enters the washer at one end and passes out at the other, being whirled along by the movement of the drum, coming in contact with the blades of the drum and the blades of the shell in its passage. Water is introduced into the washer through a series of funnels along the side, is picked up by the blades of the revolving drum, dashed into spray and thoroughly mixed with the gas. The drum is designed and built with such proportions that the tar is thrown out of the gas

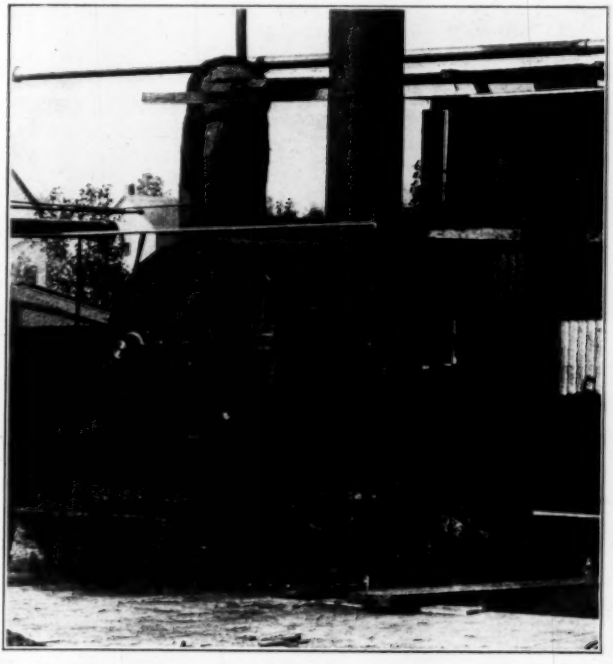

FIG. 6 SAALER ROTARY GAS WASHER, 750 H. P.

by centrifugal force. This is facilitated by the presence of the water which cools the tar from a gaseous to a liquid state and forms with it an emulsion which carries it down so it collects at the bottom of the washer while the gas passes out at the end. The washer is built with a fan attachment at the end which pulls the gas through by suction and delivers it under a slight pressure. The shell of the washer is built so as to open into an open tank filled with water,

and the tar as it collects floats on the surface of this water and then overflows a similar shield to that on the cooler.

33 This washer removes the tar from the gas in a way completely satisfactory for engine and furnace operations. The washer is sold under a warranty that the gas will contain not to exceed .015 grains of impurities all told per cubic foot. Approximately three gallons of water per h.p. hr. are used for cooling the gas in the cooler and two gal. per h.p. hr. for scrubbing the gas in the washer. The washer is self-cleansing; that is, there is no tendency of the tar to gum up the parts. It is thrown out as formed and carried off automatically

FIG. 7 INDUSTRIAL GAS PLANT

by the flow of water, so that a washer can be installed in a single unit for continuous service, and can be counted on for reliable service.

34 It is important to maintain these washers uniformly at the correct speed. If speeded too fast, the water is atomized too much and the cleansing process retarded. If speeded too slow, the centrifugal force is reduced and the tar goes through with the gas. In starting after shutdown at night, the washers require from 25 to 30 per cent overload above that required during normal operation. For this reason either a steam engine or electric generator is satisfactory for its operation.

35 This 2000 h.p. washer here under discussion is operated by 80 h.p. vertical two cylinder Westinghouse steam engine, the two being belted to a shaft.

36 The Dry Scrubber is really the final piece of apparatus connected with the gas plant proper. It is a large tank with a comparatively large diameter and small height, filled with a body of excelsior. The gas passes in at the top and out at the bottom or *vice versa* as may be most convenient, and in passing through the excelsior is freed from dust and moisture. The dry scrubber is provided with a water drain so that any moisture collecting at the bottom can be discharged. It is necessary to renew the excelsior only once every several months, the exact time varying with condi-

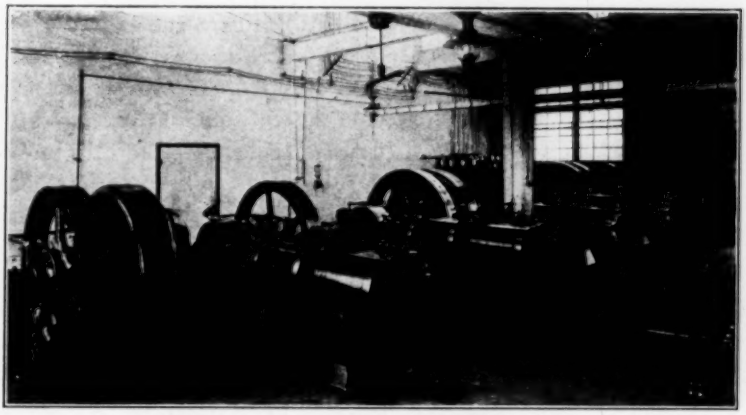

FIG. 8 ALBERGER GAS ENGINES

tions, and there is not yet sufficient data to tabulate this. The cover of this washer is removable, with a flanged rim projecting into a water seal. This makes it possible to elevate the top at a moment's notice without the handling of bolts. Each dry scrubber is accompanied by a triplex chain block and trolley on I-beam track. This is extended also over the washer and used for installing, and is on hand for any occasion where it might be necessary to make repairs. This hoisting outfit is a regular part of the gas plant equipment.

37 One of the unique features of the Industrial Gas Plant is the tar collecting and burning apparatus. This is shown in the accompanying diagrams and photographs. The tar is discharged as has

been described, the light tar passing over two steel aprons and the heavy tar discharging periodically from cocks at the bottom of the collecting tanks. Only a small amount of tar comes from the cooler, and frequently this is so mixed with dust as to be of little value, and the great bulk of the tar comes from the gas washer. As it leaves this tank it is practically a 50 per cent emulsion with water, and is barely liquid enough to flow. It drips down into the dished head of the collecting tank, and periodically when this fills up the operator lowers the valve by pushing it down and allows the tar to pass down into the tank. When this collecting tank is full, or nearly so, the operator closes the valve and turns live steam pressure

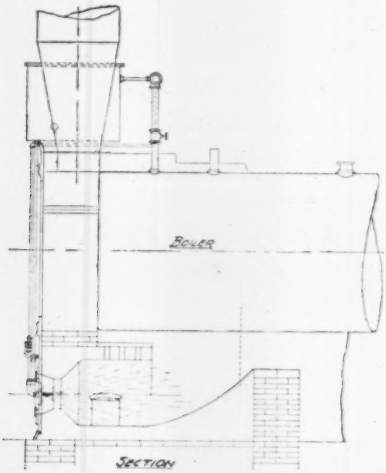

FIG. 9 TAR BURNER FIRING AUXILIARY BOILER

into the tank, which forces the tar down through the pipe going out at the bottom and up into a feed tank situated over the boiler. In connection with this 2000 h.p. plant there are two boilers and two tar feed tanks. On a 3750 h.p. installation we are now putting in, we are installing in addition to these tanks a 50-barrel tar storage tank.

38 The entire tar equipment is steam heated by coils; these steam coils are placed within the tanks and through the pipes. This keeps the tar in a liquid condition, so it can be handled and burned in the same way fuel oil is burned. The only difference is the necessity for this continual heating at all points, even through the valves, and

this necessitates somewhat bulky and expensive valves for the tar outfit. One of the diagrams shows the method of burning tar under a steam boiler. This equipment is operating in connection with twenty-four hour service, and is completely satisfactory; that is, it is satisfactory at one point where it has been finally perfected so it can be relied upon for continuous service without constant attention. In fact it requires scarcely more attention than does a fuel oil burner.

39 This equipment provides for the collection and burning of the tar, and makes it possible to use this by-product to develop steam to operate the steam engine driving the washer and steam for gas making. Complete statistics in regard to the amount of tar are not available for the reason that this apparatus has only recently been developed, and has been in service at one plant only for the past three months, but results have already established the point that the tar will furnish at least one-half the steam necessary for the operation of the gas plant.

40 We are making designs now providing for changes in connection with this gas plant whereby the exhaust steam from the engine will be used at the producers for gas making and in the tar outfit. This feature in connection with more efficient steam engines or the adoption of high speed steam turbines will, it is hoped, make it possible to develop all the steam required from the tar by-product. This, of course, will depend upon the coal, as some coals contain very little tar, but at present it looks probable that most of the Pennsylvania, Indiana and Ohio bituminous coals will give sufficient tar to provide for the operation of the plant.

41 The third Industrial Gas Plant was installed for the Garford Company, Elyria, Ohio, and was started in May, 1908. Their entire gas equipment consists of a 750 h.p. Producer Plant, 500 h.p. in Gas Engines direct connected to Generators, and a number of gas using Furnaces. Their entire equipment is piped for the use of both natural and producer gas so that either can be used, and their experience has shown that one gas can be turned on and the other turned off simultaneously without any interruption of work or without affecting operating conditions in any degree. This is true both of the engines and furnaces. The engines run full rated capacity under 115 lb. compression on producer gas, using the same compression as with natural gas. It was expected it would be necessary to raise the compression in order to get full capacity from producer gas but this did not prove to be the case, doubtless because the engines were somewhat over-rated. The engines, however, were

thoroughly adjusted at the time they were operated on producer gas, and in fact are today developing more power on producer gas than they formerly did on natural gas. This, however, is due to the better adjustment. The fact, however, is of interest as doubtless there are many plants throughout the country operating on natural gas which could be operated on producer gas without loss in capacity.

42 Very complete costs of power and of gas for furnace purposes have been made up by the Garford Company, their engineering department having run a number of tests during which they took fifteen minute readings on every possible line of data, including coal and water consumption, temperatures, pressures, switchboard readings, engine cards, gas analyses, etc. The final and official test was made during a continuous run of 125 hours made at a time when the plant was operating 24 hr. per day. These tests have established accurately the constants governing the expense at which they operate. The plant usually runs 10 hr. per day 300 days per year, carrying an average power load of 175 kw. at the switchboard, and using on the average in their furnaces 4000 ft. natural gas per hour or its equivalent. The following figures indicate the costs for a period of one year under their normal average conditions:

COST TO THE GARFORD COMPANY AT ELYRIA, OHIO, ON AN AVERAGE POWER LOAD OF 175 KW. PER HOUR, 3000 HOURS PER YEAR, AMOUNTING TO A TOTAL OF 525,000 KW. AT THE SWITCHBOARD, AND GAS USED IN FURNACES AVERAGING EQUIVALENT OF 4000 FT. NATURAL GAS PER HOUR DURING 3000 HOURS, OR A TOTAL OF 12 000 000 FT. PER YEAR

COAL

The plant ran successfully and for a long period on Ohio slack, but comparative tests determined it was cheaper to use Pittsburg run of mine on which they are now operating, the coal averaging 13 500 B.t.u. per pound, costing $1.10 at the mine with $1.00 freight and 10 cts. unloading charge, making the total cost of $2.20 per ton of 2000 lb. in the bins.

Coal burned in producer for gas for engines at 2.18 lb. per kw. hr. at switchboard, 175 kw. per hour =	383 lb.	
Coal burned in producer for furnaces requiring 85 lb. to produce the equivalent of 1000 ft., at 4000 ft., per hour	340 lb.	
Coal burned in producer for gas burned under boiler	75 lb.	
Coal burned under boiler	75 lb.	
Total coal per hour	873 lb.	
Total coal per day during working hours	8730 lb.	
Stand-by losses at night	500 lb.	
Total coal burned per day	9230 lb.	
Equals per year of 300 days, 1385 tons at $2.20		$3046.00

Steam furnished by boiler, and the coal used by boiler is divided as follows:			
Steam for Washer and Engine, 36½ h.p.		1260 lb.	
Steam for Producer		326 lb.	
Steam for Tar outfit		286 lb.	
		1872	
Supplied by 80 lb. tar	940 lb.		
Supplied by 75 lb coal burned as gas	492 lb.		
Supplied by 75 lb. coal burned on grate	440 lb.	1872	
LABOR			
One head gas man, $0.25 per hour, 4000 hours	$1000.00		
One helper, $0.18 per hour, 4000 hours	600.00		
One extra man for ash, etc., $0.18 per hour, 1500 hours	270.00	$1870.00	
One gas engine engineer at	$1500.00		
One assistant at $0.225 per hour, 3000 hours	675.00	2175.00	4045.00
WATER			
City water from Lake Erie: Engine water, 16 gal. per kw.		8,400,000	
Washer water, 1000 gal per hour		3,000,000	
Cooler, 1750 gal. per hour		5,250,000	
Boiler, 250 gal. per hour		750,000	
At $0.06 per gal.		17,400,000	1044.00
MISCELLANEOUS SUPPLIES			
Oil		$320.00	
Waste and packing		60.00	
Sparking plugs		50.00	430.00
REPAIRS			
Estimated by engineer			520.00
OVERHEAD CHARGES			
15 per cent on total cost estimated for power and fuel gas equipment of $51,620.00			7743.00
TOTAL ANNUAL COST FOR POWER AND GAS			$16828.00

SUMMARY

Items	Cost of 525 000 kw. per year	Cost of equivalent 12 000 000 ft. natural gas per year
Coal	$1635.00	$1411.00
Labor	2175.00	1870.00
Water	504.00	540.00
Miscellaneous expenses	330.00	100.00
Repairs	320.00	200.00
Overhead Charges, 15 per cent on investment of which 37½ per cent is for gas plant and 62½ per cent for engine plant	4843.00	2900.00
	$9807.00	$7021.00

Cost per kw. hour, $0.0187.
Cost for the equivalent of 1000 ft. natural gas, $0.585

44 Comparing the figures brought out by these three different installations, it may be noted that the cost of power at the switchboard is as follows for the three sets of figures given:

COST OF KW. AT SWITCHBOARD

Average load 263 h.p. or 175 kw., 10-hr. service, 300 days per year, coal at $2.20	$0.187
Average load 1320 h.p., or 892 kw., 24-hr. service, 300 days per year, coal at $2.50	.008
Average load 20 000 h.p., or 13 333 kw., 24-hr. service, 300 days per year, coal at $2.00	.004

COST OF THE EQUIVALENT OF 1000 FT. NATURAL GAS

Average use of 20 000 ft. per hr., coal at $2.00	$0.2425
Average use of 4 000 ft. per hr., coal at $2.20	0.585

45 In conclusion it may be said that the successful making of clean gas suitable for gas engines and small furnaces from bituminous coal and for continuous operation has been established beyond question; further that this type of apparatus will furnish perhaps the lowest cost of fuel to be obtained in a large proportion of the country for large power plants and for shops requiring fuel gas in large quantities. The cost of the equipment however, and of its operation, is such that under usual conditions this type of equipment is not a good investment for power plants below 250 h.p., the simple anthracite suction producer presenting a better proposition; and the advantage of this type of apparatus for furnace operation would depend

in every locality on the price at which natural gas and fuel oil can be secured and the quantity in which gas is required.

46 The figures given above show the cost under two different sets of conditions. A large plant in Pittsburg, with good gas coal at $1.25 per ton, to supply the equivalent of 250 000 to 500 000 ft. of natural gas per day, would show a cost of from $0.12 to $0.14 for the equivalent of 1000 ft. of natural gas.

NEW BOOKS

HANDBOOK FOR THE CARE AND OPERATION OF NAVAL MACHINERY. By Lieut. H. C. Dinger. *D. Van Nostrand Co., New York*, 1908. 12mo, 302 p., 124 illustrations, cloth. Price $2.50.

Contents, by parts: Operation of Naval Machinery; Care and Overhaul of Main Plant; Fittings and Auxiliaries; Care and Preservation; Sub-Division of Hull; Special Auxiliary Engines; Spare Parts and Tests.

DIE THEORIE DER WASSERTURBINEN. By Rudolf Escher. *Julius Springer, Berlin*, 1908. 8vo, 267 p., 242 figures, cloth. Price $2.70.

Contents, by chapter headings: Hydrostatik; Hydrodynamik; Die Turbinen; Die besonderen Turbinenformen; Das Regulieren; Verhalten der Turbinen unter veränderten Betriebsverhältnissen; Vergleichung der verschiedenen Bauarten; Der Spurzapfen; Die experimentelle Untersuchung.

ELEKTRISCHE BAHNEN IN NORDAMERIKA. Report of Prof. Dr. Wyssling, General Secretary of the Swiss Commission on Electric Railways. *Zurich*, 1908. 4to, 303 p., paper.

STRENGTH OF CONCRETE BEAMS. Results of Tests of 108 Beams made at the Structural Materials Testing Laboratories. By Richard L. Humphrey, U. S. Geological Survey, Washington. 1908. 8vo, 59 p., paper.

NEW EXCHANGES.

TIJDSCHRIFT VAN HET KONINKLIJK INSTITUT VAN INGENIEURS. *Gravenhaage, Holland.* Yearly. 10 by 13, paper.

ARCHIV FÜR EISENBAHNWESEN. *Berlin.* Monthly. 9½ by 6½.

DER MOTORWAGEN. *Berlin.* Weekly. 12 by 9½.

LA INGENIERIA. *Buenos Aires.* Weekly. 15 by 10½.

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THE STEAM TURBINE. By Robert Neilson. *Longmans, Green and Company, London*, 4th edition, 1908. 8vo, cloth, 604 pp., $4.20 net.

Contents, by chapter headings: Fundamental Notes and Definitions; History of the Steam Turbine; The Conversion of the Heat Energy of Steam into Kinetic Energy; Types of Steam Turbines; Losses and Efficiencies; Vane Speeds and Bucket Efficiency; Steam Turbines of Class 1; Steam Turbines of Class 2; Steam Turbines of Class 3; Steam Turbines of Class 4; Steam Turbines of Class 5; Mixed Type Steam Turbines; Low Pressure Steam Turbines; The Effects of Steam Pressure, Superheat, and Vacuum on the Efficiencies of Steam Turbines; Steam Turbine Driven Electric Generators; Steam Consumption Tests on Steam Turbines; Steam Turbine Power Stations; Ship Propulsion by Steam Turbines; Appendix 1; Equivalent Measurements; Appendix 2; British Patents for or Relating to Steam Turbines from the Earliest Records up to the end of 1905.

POWER GAS PRODUCERS. By Philip W. Robson. *Edward Arnold, London*, 1908. 8vo, cloth, 247 pp., $3.

Contents, by chapter headings: Suction Gas Producers; Pressure Plants for Non-bituminous Fuel; Pressure Producers for Bituminous Coal.

EARTHWORK AND ITS COST. By Halbert Powers Gillette, C.E., M.E. *Engineering News Publishing Company, New York*, 1907. 12mo, cloth, 244 pp., $2.

Contents, by chapter headings: The Art of Cost Estimating; Earth Shrinkage; Earth Classification; Cost of Loosening and Shoveling; Cost of Dumping, Spreading, Rolling, etc.; Cost by Wheelbarrows and Carts; Cost by Wagons; Cost by Buck and Drag Scrapers; Cost by Wheel Scrapers; Cost by the Elevating Grader; Cost by Steam Shovels; Cost by Cars; How to Handle a Steam Shovel Plant; Summary and Table of Costs; Cost of Trenching and Pipe Laying; Cost of Hydraulic Excavation; Cost of Dredging; Miscellaneous Cost Data; Earth and Earth Structures; Appendix A; Rapid Field and Office Survey Work; Appendix B; Overhaul Calculation; Appendix C; A Small "Home-Made" Dipper Dredge or Steam Shovel.

THE UTILIZATION OF WOOD WASTE BY DISTILLATION. By Walter B. Harper, M.S. *D. Van Nostrand Company, New York*, 1907. 4to, cloth, 157 pp., $3 net.

Contents, by chapter headings: Introduction; Historical Connection; Principles of Distillation; Apparatus Necessary for Destructive Distillation; Refining Methods; Special Combinations of Apparatus as used in Modern Plants; The Exaction of the Processes of Wood Distillation; Refining Process; General Consideration for the Establishment of a Plant; Composition of Wood and Products of Distillation; Yields and Disposals of Products; Chemical Tests and Combinations; Chemical Control of Plant for the Distillation of Wood; Bibliography.

CONCRETE CONSTRUCTION. By Halbert P. Gillette, C.E., M.E. and Charles S. Hill, C.E. *Myron C. Clark Publishing Company, New York and Chicago*, 1908. 8vo, cloth, 690 pp., $5 net.

Contents, by chapter headings: Methods and Cost of Selecting and Preparing Materials for Concrete; Theory and Practice of Proportioning Concrete; Methods and Costs of Making and

Placing Concrete by Hand; Methods and Cost of Making and Placing Concrete by Machine; Methods and Cost of Depositing Concrete under Water, and of Subaqueous Grouting; Methods and Cost of Making and Using Rubble and Asphaltic Concrete; Methods and Cost of Laying Concrete in Freezing Weather; Methods and Cost of Finishing Concrete Surfaces; Methods and Cost of Form Construction; Methods and Cost of Concrete Pile and Pier Construction; Methods and Cost of Heavy Concrete Work in Fortifications, Locks, Dams, Break Waters and Piers; Methods and Cost of Constructing Bridge Piers and Abutments; Methods and Cost of Constructing Retaining Walls; Methods and Cost of Constructing Concrete Foundations for Pavements; Methods and Cost of Constructing Sidewalks, Pavements, and Curb and Gutter; Methods and Cost of Lining Tunnels and Subways; Methods and Cost of Constructing Arch and Girder Bridges; Methods and Cost of Culvert Construction; Methods and Cost of Reinforced Concrete Building Construction; Method and Cost of Building Construction of Separately Molded Members; Methods and Cost of Aqueduct and Sewer Construction; Methods and Cost of Constructing Reservoirs and Tanks; Methods and Cost of Constructing Ornamental Work; Miscellaneous Methods and Costs; Methods and Cost of Waterproofing Concrete Structures.

ANALYSIS OF ELASTIC ARCHES. By Joseph W. Balet. *Engineering News Publishing Company, New York*, 1908. 8vo, cloth, 316 pp., $3.

Contents, by chapter headings: The Arch Rib; Three Hinged Arches; The Two Hinged Arch; Hingeless Arches; Stresses in Arch Sections; Stresses Caused by Lateral Wind Pressure; Appendix, Notation; Mathematical Analysis of the Elastic Arch; Arched Frame Works.

WOOD. A Manual of the Natural History and Industrial Applications of the Timbers of Commerce. By G. S. Boulger, F.L.S., F.G.S., F.R.H.S., A.S.I. *Edward Arnold, London*, 2d edition, 1908. 8vo, cloth, 348 pp., $4.20.

Contents, by chapter headings: Part 1—Of Wood in General; The Origin, Structure and Development of Wood and its Use to the Tree; The Recognition and Classification of Wood; Defects of Wood; Selection, Durability, Seasoning and Storage of Woods; The Use of Woods; Our Supplies of Wood; Testing Wood. Part 2—Woods of Commerce, Their Sources, Characters and Uses; Appendices; Explanation of Some Terms Used with Reference to Converted Timber, etc.; The Microscopic Examination of Woods; Select Bibliography; The Distinctive Microscopic Structures of Woods.

ECONOMICS OF RAILWAY OPERATION. By M. L. Byers, C.E. *Engineering News Publishing Company, New York*, 1908. 8vo, cloth, 672 pp., $5.

Contents, by chapter headings: Organization, Employment, Education and Discipline of Forces; Accounts and Accounting; Reports; Economic Operation; Analysis of Operations and Control of Expenses; Betterments.

RAILWAY TRACK AND TRACK WORK. By E. E. Russel Tratman, C.E. *Engineering News Publishing Company, New York*, 3d Edition, Fully Revised, 1908. 8vo, cloth, 520 pp., $3.50.

Contents, by chapter headings: Part 1—Track: Introduction; Roadbed Construction; Ballast; Ties and Tie-Plates; Rails; Rail Fastenings and Rail Joints; Switches, Frogs and Switchstands; Fences and Cattle-Guards; Grade Crossings; Bridge and Trestle Floors; Track Signs; Track Accessories; Sidings; Yards and Terminals; Track Tools and Supplies; Signals and Interlocking; Electric Railways. Part 2—Track Work: Organization of the Maintenance-of-Way Department; Tracklaying; Ballasting and Renewing Rails; Drainage and Ditching; Track Work for Maintenance; Gage, Grades and Curbs; Track Inspection and the Premium System; Switch Work and Turnouts; Bridge Work and Telegraph Work; Permanent Improvements; Handling and Clearing Snow; Wrecking Trains and Operations; Records, Reports and Accounts; Appendix.

STRESSES IN STRUCTURES. By A. H. Heller, C.E. *John Wiley and Sons, New York*, 1908. 8vo, cloth, 324 pp., $3.

Contents, by chapter headings: Stresses and Deformations within the Elastic Limits; Stresses and Deformations beyond the Elastic Limit; The Laws of Equilibrium and Their Application; Application of theLaws of Equilibrium to the Structures as a Whole—Reactions; Application of the Laws of Equilibrium to any One Joint; Application of the Laws of Equilibrium to any Part of a Structure; Stresses in Beams and Girders; Deflections of Beams and Girders; Special Cases of Beams and Girders Loaded and Supported in Different Ways; Stresses in Blocks and Columns; Types of Trusses; Stresses in Simple Bridge Trusses for Uniform Loads; Stresses in Railway Bridges from Wheel Loads; Stresses in Bridges from Horizontal Forces.

LOCOMOTIVE ENGINE RUNNING AND MANAGEMENT. By Angus Sinclair, M.E. *John Wiley and Sons, New York*, 22d Edition, Enlarged, 1908. 12mo, cloth, 442 pp., $2.

Contents, by chapter headings: Introduction; Engineers and their Duties; How Engineers are Made; Inspection of the Locomotive; Getting Ready for the Road; Running a Fast Freight Train; Getting up the Hill; Finishing the Trip; Hard Steaming Engines; Shortness of Water; Boilers and Fire Boxes; Accidents to the Valve Motion; Accidents to Cylinders and Steam Con nection; Off the Track; Accidents to Running Gear; Connecting Rods, Side Rods and Wedges; Valve Motion; The Shifting Link; Setting the Valve; The Westinghouse Air Brake; Tractive Power and Train Resistance; Draft Appliances; Combustion; Steam and Motive Power; Sight Feed Lubricators; Examination of Firemen for Promotion.

THE MARINE STEAM TURBINE. By L. W. Sothern. *D. Van Nostrand Company*, 2d edition, 1906. 8vo, cloth, 163 pp., $2.50.

Contents, by chapter headings: Definitions and General Principles; Practical Construction, etc.; Data from Practice, Propellers, Consumption, etc.

THE STEAM ENGINE AND OTHER STEAM MOTORS. By Robert C. H. Peck, M.E. *D. Van Nostrand Company*, 2d Edition, Revised, 1907. 8vo, cloth, 391 + 678 pp. In 2 vols. Vol. 1, $5, The Thermodynamics and The Mechanics of the Engine; vol. 2, $3.50, Form, Construction and Working of the Engine; The Steam Turbine.

AERODYNAMICS. By F. W. Lanchester. *D. Van Nostrand Company, New York*, 1908. 8vo, cloth, 442 pp., $6.

Contents, by chapter headings: Fluid Resistance, and Associated Phenomena; Viscosity and Skin Friction; The Hydrodynamics of Analytical Theory; Wing Form and Motion in the Periptery, The Aeroplane; The Normal Plane; The Inclined Aeroplane; The Economics of Flight; The Aerofoil; On Propulsion; the Screw Propeller, and the Power Expended in Flight; Experimental Aerodynamics.

INDUSTRIAL ALCOHOL. By John Geddes McIntosh. *Scott, Greenwood and Son, London*, 1907. 8vo, cloth, 252 pp., $3.

Contents, by chapter headings: Alcohol and its Properties; Continuous Fermentation and Sterilization in Industrial Alcohol; Manufacture; The Manufacture of Industrial Alcohol from Beets; The Manufacture of Industrial Alcohol from Grain; The Manufacture of Industrial Alcohol from Potatoes; The Manufacture of Industrial Alcohol from Surplus Stocks of Wine; The Manufacture of Alcohol from the Sugar Cane and Sugar Cane Molasses; Plant, etc., for the Distillation and Rectification of Industrial Alcohol; The Manufacture and Uses of Various Alcohol Derivatives; The Uses of Alcohol in Manufactures, etc.; The Uses of Alcohol for Lighting, Heating and Motive Power.

AIRSHIPS, PAST AND PRESENT. By A. Hildebrandt. *D. Van Nostrand Company, New York*, 1908. 8vo, cloth, 361 pp., $3.50.

Contents, by chapter headings: Early History of the Art; Invention of the Air Balloon; Montgolfières, Charlières, and Rozières; The Theory of the Balloon; The Development of the Dirigible Balloon; The History of the Dirigible Balloon, 1852–1872. 1883–1897; Dirigible Balloons from 1898–1906; Flying Machines; Kites; Parachutes; The Development of Military Ballooning; Ballooning in the Franco-Prussian War; Modern Organization of Military Ballooning in France, Germany, England and Russia; Military Ballooning in other Countries; Balloon Construction and the Preparation of the Gas; Instruments; Ballooning as a Sport; Scientific Ballooning; Balloon Photography; Photographic Outfit for Balloon Work; The Interpretation of Photographs; Photography by Means of Kites and Rockets; Problems in Perspective; Carrier Pigeons for Balloons; Balloon Law.

EQUIPMENT AND OPERATION OF THE SUBWAY OF THE INTERBOROUGH RAPID TRANSIT COMPANY OF NEW YORK CITY. Submitted to the Public Service Commission of the State of New York. By Bion J. Arnold, Special Consulting Engineer. *Report No. 1. New York*, 1907. 8vo, paper, 9 pp.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

021 "Wanted—Capable Engineer" to sell condensers. Philadelphia concern.

022 Electrical Engineer along laboratory and design to fill position for one year on an acting Assistant Professorship at $1600. University in the Middle West.

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025 An Eastern firm wishes a number of experienced designers of direct acting, power, express, mine and multistage centrifugal pumps. Location Pennsylvania.

026 Wanted by a Canadian firm of contractors, a young man who thoroughly understands heating and ventilating work, ability to make estimates and consult with architects.

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134 Member; operating engineer of well known ability and reputation, patentee of various apparatus, seven years with large industrial concern as steam and mechanical expert, recently conducting consulting office, will consider early engagement as supervising engineer of power plants or chief engineer of large plant.

135 M.E., Stevens, 1898. Ten years experience power plant construction, factory equipment and maintenance, thoroughly acquainted with up to date methods of factory management and accounting. Position as superintendent of maintenance, construction, or as mechanical engineer with contracting firm.

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137 Junior desires position as superintendent, or mechanical engineer; five years experience designing, erecting, and building Corliss engines, condensers, pumps, heaters, etc. Two years designing, and estimating power plant work. Shop and business experience, executive ability.

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DEATHS

Frank B. Kleinhans, Francis X. McGowan.

COMING MEETINGS OF SOCIETIES

AMERICAN ELECTROCHEMICAL SOCIETY

October 30, College of the City of New York. Secy., Dr. J. W. Richards, South Bethlehem, Pa. Acting Secy., W. S. Landis.

AMERICAN GAS INSTITUTE

October 2–23, annual convention, 29 W. 39th St., New York. Secy., James W. Dunbar, New Albany, Ind.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

October 9, 33 W. 39th St., New York, 8 p. m. Secy., R. W. Pope. Paper: High Potential Underground Transmission, P. Junkersfeld, E. O. Schweitzer.

AMERICAN SOCIETY OF CIVIL ENGINEERS

Oct. 7, New York, Secy., Chas. W. Hunt, 220 W. 57th St.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

October 13, monthly, December 1–4, annual meeting, 29 West 39th Street, New York. Secretary, Calvin W. Riee.

AMERICAN SOCIETY OF MUNICIPAL IMPROVEMENTS

October 20–23, annual convention, Atlantic City, N. J. Secy., A. Prescott Folwell, Flatiron Building, New York.

AMERICAN STREET AND INTERURBAN RAILWAY ASSOCIATION

October 12–16, annual meeting at Atlantic City, N. J. Secy., Bernard V. Swenson, 29 West 39th St., New York.

ASSOCIATION OF RAILWAY SUPERINTENDENTS OF BRIDGES AND BUILDINGS

October 20–22, annual meeting, Arlington Hotel, Washington, D. C. Secy., S. F. Patterson, Concord, N. H. Papers: Water-proofing of Concrete Covered Steel Floors; Modern Equipment and Tools for Erection of Steel Bridges; Protection of Structures Against Effects of Electric Currents; Protection of Embankments from the Effects of High Water by Rip-rap or Otherwise; Experience in the Use of Gasoline Engines and Kerosene Engines or Combination of Same for Water Supply, Drawbridges, Etc.; Modern Dwelling Houses for Section Foremen and Section Men in Outlying Districts; Re-enforced Concrete Culverts and Short Span Bridges; Methods of Erecting Truss Bridges, *a*, Maintaining Traffic; *b*, No Traffic; Smoke Jacks for Engine Houses.

BOSTON SOCIETY OF CIVIL ENGINEERS

October 21, monthly meeting, Tremont Temple. Secy., S. E. Tinkham, 60 City Hall.

BROOKLYN ENGINEERS' CLUB

October 8, 197 Montague St. Secy., J. Strachan. Paper: Some Features of the Bridges of the Harlem River Branch of the N. Y., N. H., & H. R. R., Herbert C. Keith.

CANADIAN FREIGHT ASSOCIATION

October 8, Montreal, Que. April 9, 1909, annual meeting. Secy., T. Marshall, Toronto, Ont.

CANADIAN RAILWAY CLUB

October 6, Montreal, Que. Secy., James Powell, Grand Trunk Railway, Montreal.

CANADIAN SOCIETY OF CIVIL ENGINEERS

Oct. 8, Montreal, Que. Secy., Clement H. McLeod, 413 Dorchester St., W.

CAR FOREMENS' ASSOCIATION OF CHICAGO

October 12. Secy., Aaron Kline, 326 N. 50th St.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS

October 5, Indianapolis, Ind. January 1, 1909, annual meeting, 257 Broadway. Secy., G. B. Staats, Care of Penna. Lines.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS

October 12, Kansas City, Mo. Secy., J. H. Ashley, Gumbel Bldg.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS

October 8, Toledo, O. Secy., H. M. Elliott.

CENTRAL RAILWAY AND ENGINEERING CLUB OF CANADA

October 20, Rossin House, Toronto, Ont. Secy., C. L. Worth, Room 409, Union Station. Paper: The Handling of Stores, A. Tory.

CLEVELAND ENGINEERING SOCIETY

October 13, monthly meeting, Caxton Building. Secy., Joe. C. Beardsley.

COLORADO SCIENTIFIC SOCIETY

October 3, monthly meeting, Chamber of Commerce Building, Denver. December 19, annual meeting. Secy., Dr. W. A. Johnston, 801 Symes Bldg.

EMPIRE STATE GAS AND ELECTRIC ASSOCIATION

October 7, annual meeting at New York. Secy., C. H. B. Chapin, 154 Nassau St. Papers on taxation, insurance, public policy of the association.

ENGINEERING ASSOCIATION OF THE SOUTH

October 20, monthly meeting, Nashville Section, Carnegie Library Bldg. Section Secy., H. H. Trabue, Berry Block, Nashville.

ENGINEERING SOCIETY OF THE STATE UNIVERSITY OF IOWA

October 6, monthly meeting. Dean, Wm. G. Raymond.

ENGINEERS AND ARCHITECTS' CLUB OF LOUISVILLE, KENTUCKY

October 19, monthly meeting, 303 Norton Bldg. Secy., Pierce Butler.

ENGINEERS' CLUB OF BALTIMORE

October 3. Secy., R. K. Compton, City Hall.

ENGINEERS' CLUB OF CENTRAL PENNSYLVANIA

October 6, monthly meeting, Gilbert Bldg., Harrisburg. January 5, 1909, annual meeting. Secy., E. R. Dasher.

ENGINEERS' CLUB OF CINCINNATI

October 15, 25 E. 8th St. Secy., E. A. Gast, P. O. Box 333.

ENGINEERS' SOCIETY OF MILWAUKEE

October 14, monthly meeting; June 9, annual meeting. Secy., W. Fay Martin, 396 Bradford Ave.

GENERAL SUPERINTENDENTS' ASSOCIATION

October 14, Chicago, Ill. Secy., H. D. Judson, C. B. & Q. R. R., Chicago.

ILLUMINATING ENGINEERING SOCIETY

October 6–7, annual convention at Philadelphia, Pa. Secy., Van Rensselaer Lansingh, 33 West 39th St., New York.

IOWA RAILWAY CLUB

October 9, Des Moines. Secy., W. B. Harrison, Union Sta.

KANSAS GAS, WATER AND ELECTRIC ASSOCIATION

October 8–9, annual meeting at Pittsburgh. Secy., J. D. Nicholson.

LOUISIANA ENGINEERING SOCIETY

October 12, monthly meeting, 323 Hibernia Bldg., New Orleans, 8 p. m. January 9, 1909, annual meeting, New Orleans. Secy. L. C. Datz, 323 Hibernia Bldg.

MASSACHUSETTS STREET RAILWAY ASSOCIATION

October 14, Boston. Secy., Charles S. Clark, 70 Kilby St.

NATIONAL MACHINE TOOL BUILDERS' ASSOCIATION

October 20–21, annual convention, Hotel Imperial, New York. Secy., P. E. Montanus, Springfield, O.

NATIONAL RAILWAY WATER SUPPLY ASSOCIATION

October 11, Minneapolis, Minn. Secy., F. W. Hayden, Glencoe.

NEW ENGLAND RAILROAD CLUB

October 13, monthly meeting at Boston, Mass. Secy., Geo. H. Frazier, 10 Oliver St., Boston.

NEW YORK ELECTRICAL SOCIETY

Second week in October. Secy., Geo. H. Guy, 29 W. 39th St., New York.

NEW YORK RAILROAD CLUB

October 16, monthly meeting, 29 West 39th St. Secy., H. D. Vought, 62 Liberty St.

NORTHWEST RAILWAY CLUB

October 13, Minneapolis, Minn. Secy., T. W. Flannagan, Soo Line, Minneapolis.

PACIFIC NORTHWEST SOCIETY OF ENGINEERS

October 3, monthly meeting, Lowman Bldg., Seattle, Wash. Secy., Arthur H. Dimock, City Engineer's Office. Paper: The Possibilities of the Hydroplane, L. E. Meacham.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS

October 27, monthly meeting, 48 Snow St. June 22, 1909, annual meeting. Secy., T. M. Phetteplace.

RAILWAY CLUB OF PITTSBURGH

October 23, annual meeting, Monongahela House, 8 p.m. Secy., J. D. Conway, Gen. Office, P. & L. E. R. R.

RAILWAY SIGNAL ASSOCIATION

October 13–15, annual meeting at Washington, D. C. Secy., C. C. Rosenberg, 712 North Linden St., Bethlehem, Pa.

RICHMOND RAILWAY CLUB

October 12. Secy., F. O. Robinson, C. & O. Railway.

ROCHESTER ENGINEERING SOCIETY

October 9, monthly meeting. Secy., John F. Skinner, 54 City Hall.

SHORT LINE RAILROAD ASSOCIATION

October 5, New York. Secy., Cromwell G. Macy, Jr., Nantucket Central Railroad, 257 Broadway.

ST. LOUIS RAILWAY CLUB

October 9, monthly meeting. Secy., B. W. Frauenthal.

TECHNICAL SOCIETY OF BROOKLYN

October 16, semi-monthly meeting, Arion Hall, Arion Place, Brooklyn, N. Y. 8:30 p. m. Pres., M. C. Budell, 20 Nassau St., New York.

TECHNOLOGY CLUB OF SYRACUSE

October 13, monthly meeting, 502 The Bastable Blk. Secy., Geo. D. Babcock.

WESTERN RAILWAY CLUB

October 20, monthly meeting, Auditorium Hotel, Chicago, Ill., 8 p.m. Secy., Jos. W. Taylor, 390 Old Colony Bldg.

WESTERN SOCIETY OF ENGINEERS

October 7, 21, bi-weekly meetings, 1737 Monadnock Blk., Chicago, Ill., January 5, 1909, annual meeting. Secy., J. H. Warder. Papers: October 7, Methods of Studying the Heat-Absorbing Properties of Steam Boilers; October 21, Notes on Macadam Road Construction, A. N. Johnson.

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